Stirling

A Book on Steam for Engineers

ENGRAVED AND PRINTED BY
ROSENOW COMPANY, CHICAGO

The Stirling Company

Manufacturers of

Water-Tube Safety Boilers for Stationary and Marine Use,

Superheaters, Bagasse Furnaces and Conveyors,

Chain Grate Stokers, Steel Stacks,

and Breechings

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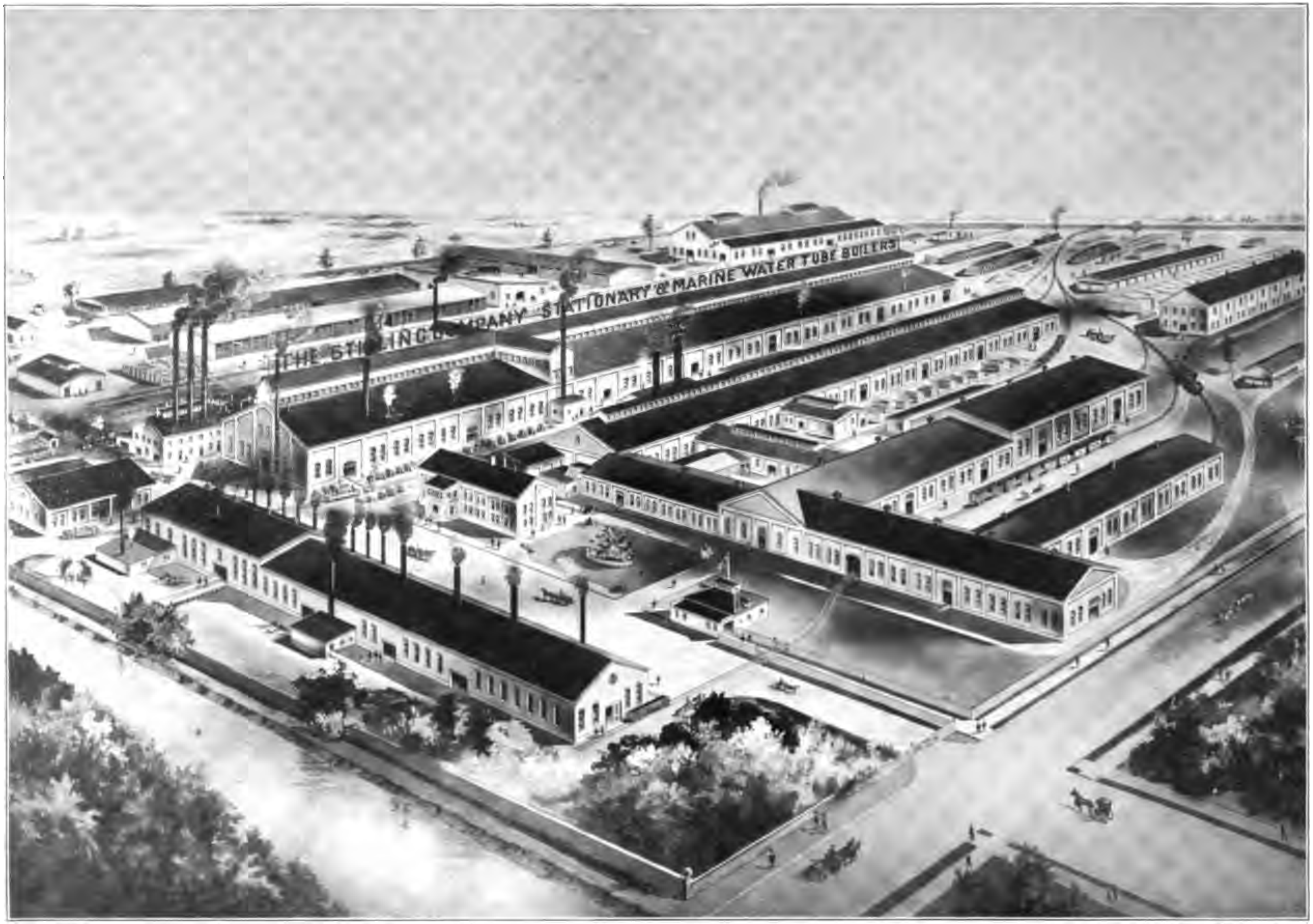
Pub. No. 1205



WORKS OF THE STIRLING COMPANY, BARBERTON, OHIO

Stirling

A Book on Steam for Engineers

Edited by

The Engineering Staff of The Stirling Company

New York

The Stirling Company

Trinity Building

1905

Special Notice

Realizing that it is practically impossible to avoid errors or misprints in the first edition of a work of this size, the publishers cordially invite those who note errors of any kind to report them, so that the necessary corrections may be made in future editions

THE STIRLING COMPANY

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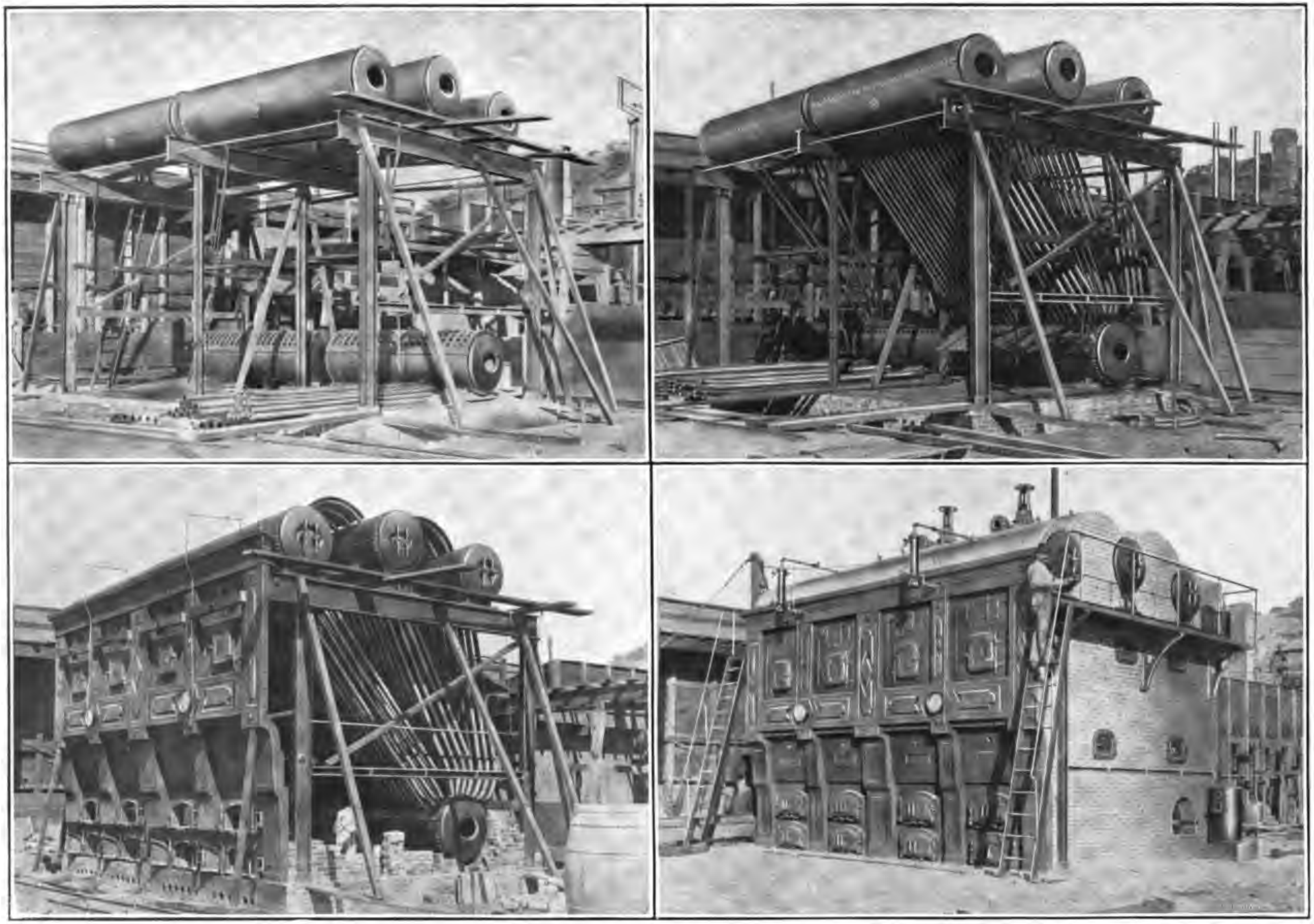


ERECTING STIRLING BOILERS AT SELBY SMELTING & LEAD WORKS, VALLEJO JUNCTION, CAL. 900 H. P. IN OPERATION

The Stirling Water-Tube Safety Boiler

Shortly after the invention of the double-acting steam engine by Watt, the development of the water-tube boiler began, but of the many designs which appeared during the succeeding half century none proved successful. Yet not all of the ideas underlying these early and crude designs were valueless, and after a long course of experimenting some of them were gradually worked out to a practical application, and the water-tube boiler then became a commercially successful steam generator. It was far from being perfect, however, as many of the most advanced ideas of inventors could not be embodied in these early types of boilers, because of the lack of suitable material to construct the parts, and the then inadequate mechanical facilities for building such boilers. In consequence of these manufacturing limitations, the general design narrowed down to some arrangement in which steam was generated in slightly inclined tubes, and discharged into one or more upper drums. These parts were all easily made with the materials and mechanical facilities available at that time. One almost insuperable difficulty remained, and that was to devise safe and efficient means of providing passageway from the tube ends into the drums. Out of the hundreds of methods tried, but few proved to be even approximately successful, and there is yet to be found a method which can be considered entirely satisfactory. In consequence of the complexity of parts required in these various designs for connecting the tubes and the drums, cast iron was the only available material, and very unfortunately, its use for making such parts became common.

The tendency to follow in the beaten track is well illustrated in this case, because until about two decades ago practically all of the various water-tube boilers which had achieved any success were more or less complicated developments of the general scheme of attaching nearly horizontal tubes to a drum. In consequence, all of them were distinguished by a multitude of joints, caps, bolts, headers, water-legs, nipples, and other objectionable features, while practically all of them were compelled to use cast iron in headers, return-bends, and other parts of complicated shape subjected to high pressure. As steam pressures and the sizes of boilers were gradually increased, the defects of this general design were found to be many, and as each defect developed, further complication was introduced to correct it.

These complications being inherent in the general design, it follows that their elimination demanded the development of a new type of boiler so essentially different, that, without introducing any new defects, it would be free from those affecting the older types whose possibilities of development had been exhausted. The problem of producing such a boiler received the earnest attention of many engineers, yet there was, and is now, but one satisfactory solution for that problem, the STIRLING WATER-TUBE SAFETY BOILER, as now manufactured and offered by THE STIRLING COMPANY.

Every great invention is the result of gradual evolution, and the Stirling boiler is no exception to this law. The first boilers of this type contained one mud drum and only two steam drums. These boilers were crudely constructed, and in their installation but little attention was paid to those minor details the aggregate of which constitute perfection. Crude, however, as these first boilers were, they conclusively demonstrated that the principle of the boiler is correct and that great possibilities lay in the development of its application. These points having been established, THE STIRLING COMPANY was formed, the boiler was developed, and its construction was perfected, but its principle was and always has been, the same. In its improved form, as described in the following pages, it has met every demand, and fulfilled every requirement.

The Stirling Boiler (Figs. 1 and 2) consists of three upper or steam drums, each connected by a number of tubes (called a "bank") to a lower or mud drum. Suitably disposed firetile baffles between the banks direct the gases into their proper course. Shorter tubes connect the steam spaces of all

upper drums, also water spaces of front and middle drums. The boiler is supported on a structural steel framework, around which is built a brick setting whose only office is to provide furnace space, and serve as a

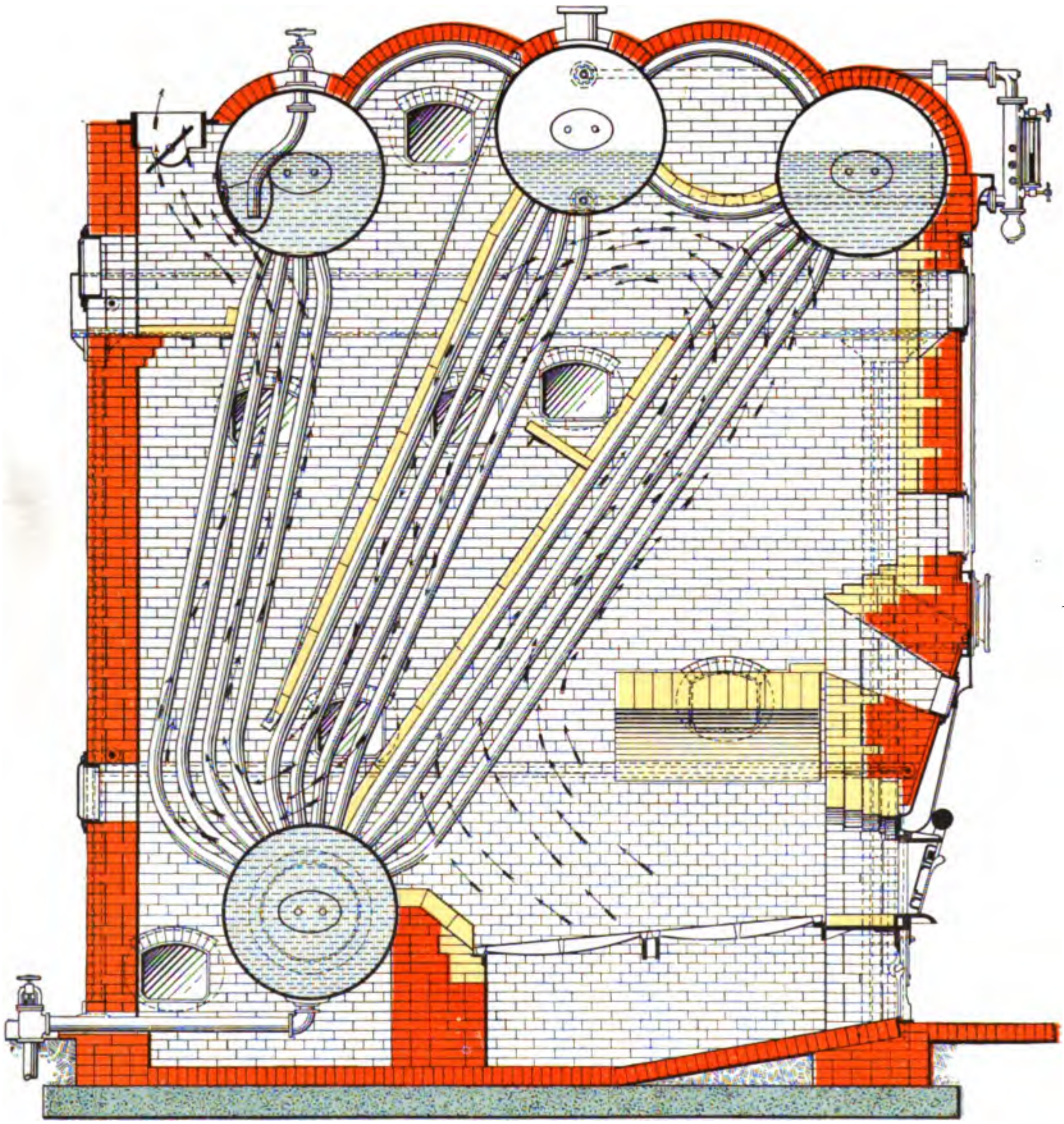


FIG. 1. THE STIRLING WATER-TUBE SAFETY BOILER—SECTIONAL SIDE ELEVATION
THE RED, YELLOW AND BLUE SECTIONS RESPECTIVELY INDICATE RED BRICK, FIRE-BRICK, AND CONCRETE

housing to confine the heat. The entire front is of metal of appropriate and artistic design. These parts, together with the usual valves and fittings, constitute the completed boiler, which represents the acme of simplicity and eliminates the complication of the older types.

The Drums vary from 36 to 54 inches in diameter and are made of the best open hearth flange steel. The plates extend the entire distance between heads, hence there are *no circular seams*. The longitudinal seams—which are double or triple riveted according to the working pressure to be carried—are so placed that they are not exposed

to high temperature. The drum heads are hydraulically dished to proper radius; each drum is provided with one manhole, and the manhole plate and arch bars are of wrought steel; four manhole plates, which can be removed in ten minutes, give access to the entire interior of the boiler, and expose every tube end, rivet, and joint. The drum interiors are perfectly clear; there are no baffles, stays, tie-rods, mud pipes, or other obstructions in them, as evidenced by the sectional view shown in Fig. 3.

The Tubes are best lap-welded mild steel. They are slightly curved at the ends to permit them to enter the drums normally and to provide for free expansion of the boiler when at work. The tubes are expanded directly into reamed holes in tube sheets of the drums, hence the annular recess between tubes and the cast headers of some types of boiler is

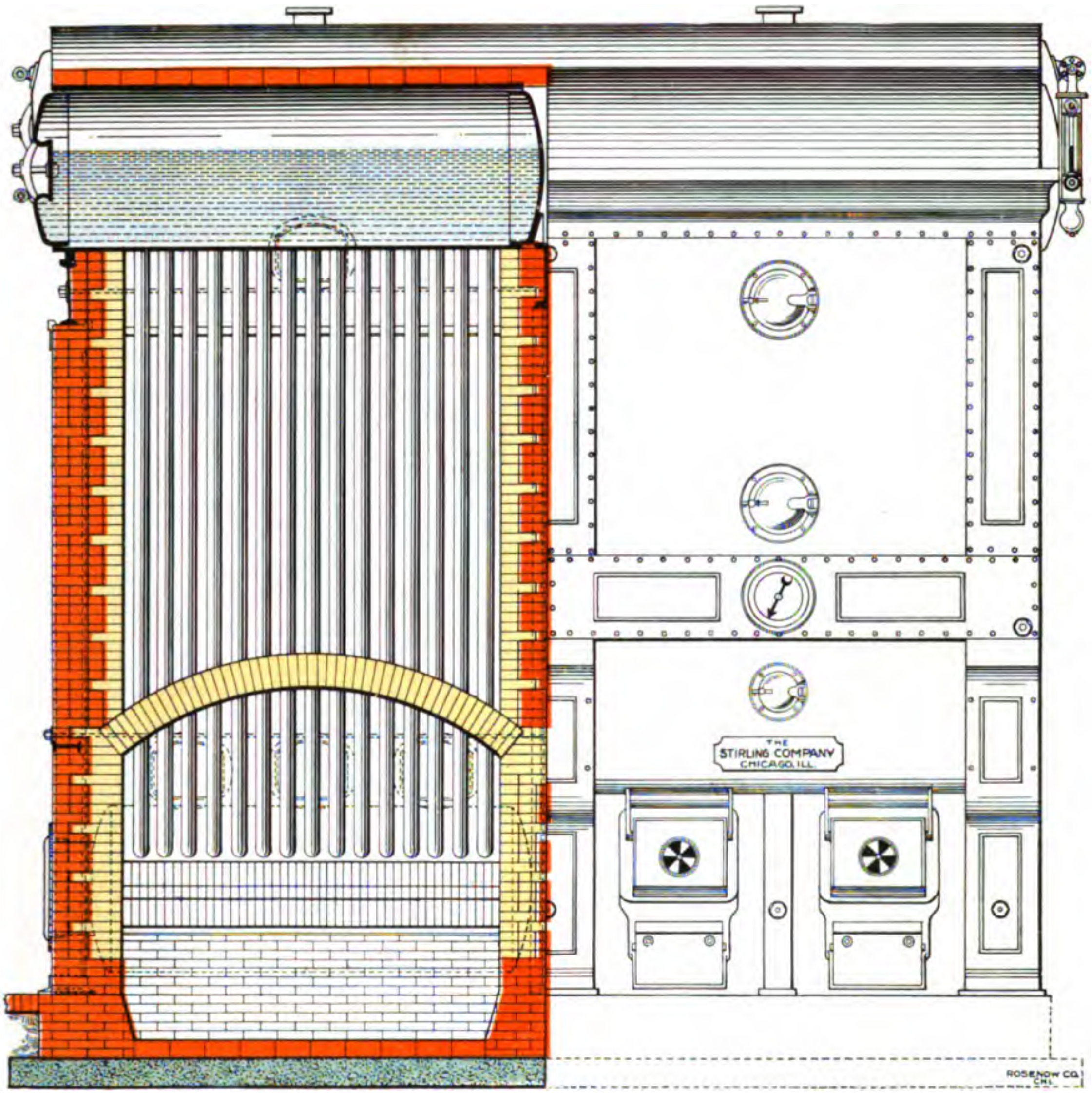


FIG. 2. THE STIRLING WATER-TUBE SAFETY BOILER—SECTIONAL FRONT ELEVATION

THE RED, YELLOW AND BLUE SECTIONS RESPECTIVELY INDICATE RED BRICK, FIRE-BRICK, AND CONCRETE

eliminated, and failure of tubes by pitting through corrosion caused by accumulation of soot in these recesses is avoided. There are no short nipples and no tube joints in places which can be reached only by jointed handles on the tube expander, rendering it impossible to determine when the tube has been properly expanded. In the Stirling every tube end is visible and accessible.

Steel Framework—As the entire weight of boiler and contents is supported on the steel framework, cracking of the setting due to unequal settlements is obviated, and no blocking is needed when the brickwork has to be repaired. The design of framework can be modified to suit special conditions.*

Furnace—The design of the Stirling furnace is a distinct advance over previous practise, and offers advantages wholly unobtainable under some types of boiler, and obtainable under the others only by a prohibitive increase in floor space. Referring to Figs 1 and 2, it will be seen that a fire-brick arch is sprung over the grates and immediately in front of the first bank of tubes. The large triangular space between boiler front, tubes, and mud drum, is available for combustion chamber, and for installation of sufficient grate surface to meet the requirements of the lowest grades of fuel, all in marked contrast to boilers of the internally fired type, and many water-tube boilers in

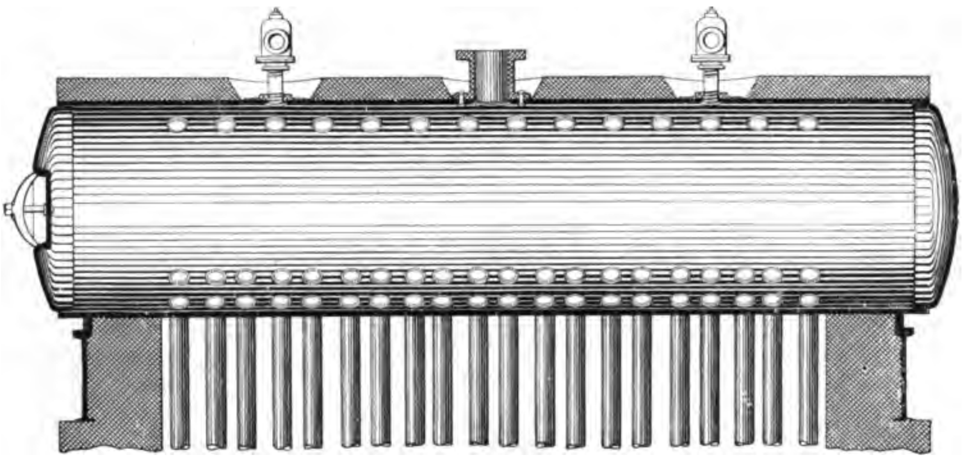


FIG. 3. SECTION THROUGH STEAM DRUM, SHOWING ABSENCE OF BAFFLES OR OTHER COMPLICATIONS

which only the same grate surface is available whether the vertical rows contain few or many tubes.

Brick Setting—This is so clearly shown by the cuts (Figs. 1 and 2) that extended description is unnecessary. It is all plain, straight work, which can be done by any good brick mason who can read lucid instructions and a simple drawing. No special shapes or other material not found in open market are needed. Any necessary repairs to brickwork can be made without disturbing the boiler connections. The setting is provided with numerous doors of ornamental design (Fig. 4), which give access to all parts for cleaning.

Kent† says: "Coal can be burned without smoke, provided:

(I) "The gases are distilled from the coal *slowly*.

(II) "That the gases when distilled are brought into intimate contact with *very hot* air.

(III) "That they are burned in a *hot fire-brick chamber*.

(IV) "That while burning they are not allowed to come into contact with compara-

*As an evidence of the direct advantages resulting from this manner of supporting the boiler, we refer to an explosion of natural gas in the furnace of a Stirling Boiler at the American Tin Plate Company's plant at Elwood, Ind. Although the force of the explosion was sufficient entirely to demolish the brickwork, the boiler was uninjured in any way whatever. The brick was replaced and the boiler put into commission, without its having been necessary to make any repairs whatsoever to the boiler proper. A recent similar explosion under a boiler fired with oil at Works of Santa Monica, (Cal.) Brick & Tile Mfg. Co. showed precisely the same result—brickwork completely demolished, but boiler uninjured. †*Steam Boiler Economy*, First Edition, p. 156.

tively cool surfaces, such as the shell or tubes of a steam boiler; this means that the gases shall have sufficient space and time in which to burn before they come into contact with the boiler surfaces.''

The first condition demands careful firing, and sufficient grate surface; this grate surface is available in the Stirling furnace.

FIG. 4. CLEANING DOORS WITH ASBESTOS PACKING WASHERS

The second requirement is preeminently met by introduction of the brick arch, which absorbs heat from the fire, becomes an incandescent radiating surface similar to roof of a reverberatory furnace, heats up any air admitted over the fuel, and ignites by radiation the gases distilled from the coal; it insures an even distribution of the gases, obviates their concentration at any one point and prevents the boiler from being chilled by inrush of cold air when the furnace doors are opened.

The third requirement is met because the arch in combination with the furnace walls forms a fire-brick chamber of large capacity.

The fourth requirement cannot be met by any internally-fired boiler, or water-tube boiler in which tubes form the roof of the furnace. In the Stirling furnace the gases *do not come into contact with tubes* until they pass out of the fire-brick chamber under the arch, and this chamber is of sufficient size to allow the gases space and time in which to burn.

Baffles and Course of Gases—The baffle walls rest directly upon the tubes, and guide the course of the gases up the front bank, down the middle and up the rear bank, thus bringing them into such intimate contact with the boiler surface that the heat is quickly and thoroughly extracted from them. In no other boiler are the gases compelled to travel as far before reaching the stack, and the effect upon economy is evident. The baffles are made of plain rectangular firetile carried in stock by all fire-brick dealers, in contrast to the special formed bricks (obtainable only from the manufacturer) required by many types of water-tube boiler. Another marked advantage of the Stirling baffles is that since no tubes pass *between* or through the tiles (see Fig. 5), they are not pried apart and made leaky by distorted tubes; they can be removed and replaced without disturbing a tube. Baffles built across the tubes, as in many boilers, are damaged by pulling a faulty tube through them, and can be repaired in but one way—by removal of every tube necessary to permit a man to crawl in and reach the defective spot.

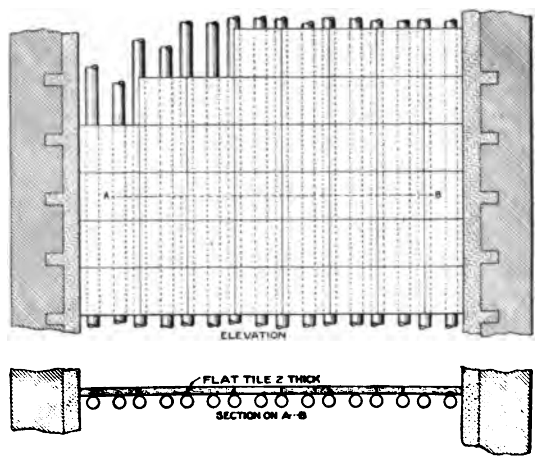


FIG. 5. ELEVATION AND SECTION OF FIRETILE BAFFLES IN STIRLING BOILERS

ADVANTAGES OF THE STIRLING CONSTRUCTION.

Such advantages as have not already been named will now be pointed out. When comparisons are made with other types, they are given not with the intention of attacking or disparaging those types, but merely to

WORKS OF COLORADO FUEL & IRON CO., PUEBLO, COLORADO

OPERATING 29,000 H. P. OF STIRLING BOILERS. UPPER VIEWS REPRESENT 18,000 H. P. IN COURSE OF ERECTION

Photographs copyrighted, 1902, by G. M. Laybourn, Pueblo, Colo.

bring out the superior points of the Stirling water-tube safety boiler.

Simplicity — There are no details of complicated shape; no flat surfaces, tie-rods, water-legs, headers, return-bends, outside circulating pipes to plug up; no multitudinous handhole plates to be removed and packed with gaskets, or be ground and scraped to a fit whenever boiler is opened; no baffles or mud pipes in the drums; no short nipples, seams exposed to heat, or parts inaccessible for cleaning.

Expansion and Contraction—In the Stirling the mud drum is not embedded in brickwork, but is suspended on the tubes which connect it with the upper drums. While in many water-tube boilers the weight of all the tubes and heavy headers must be supported by a single row of nipples in front and another row at the rear of the boiler—which nipples frequently work loose owing to the vibrations ever present in a boiler when at work—the method of suspension in the Stirling is radically different; here the mud drum is suspended on *all* the long tubes and the weight carried by each tube in supporting the drum and its contained water is only about forty pounds. Besides this the tubes are curved, so that each one may independently of the others expand or contract

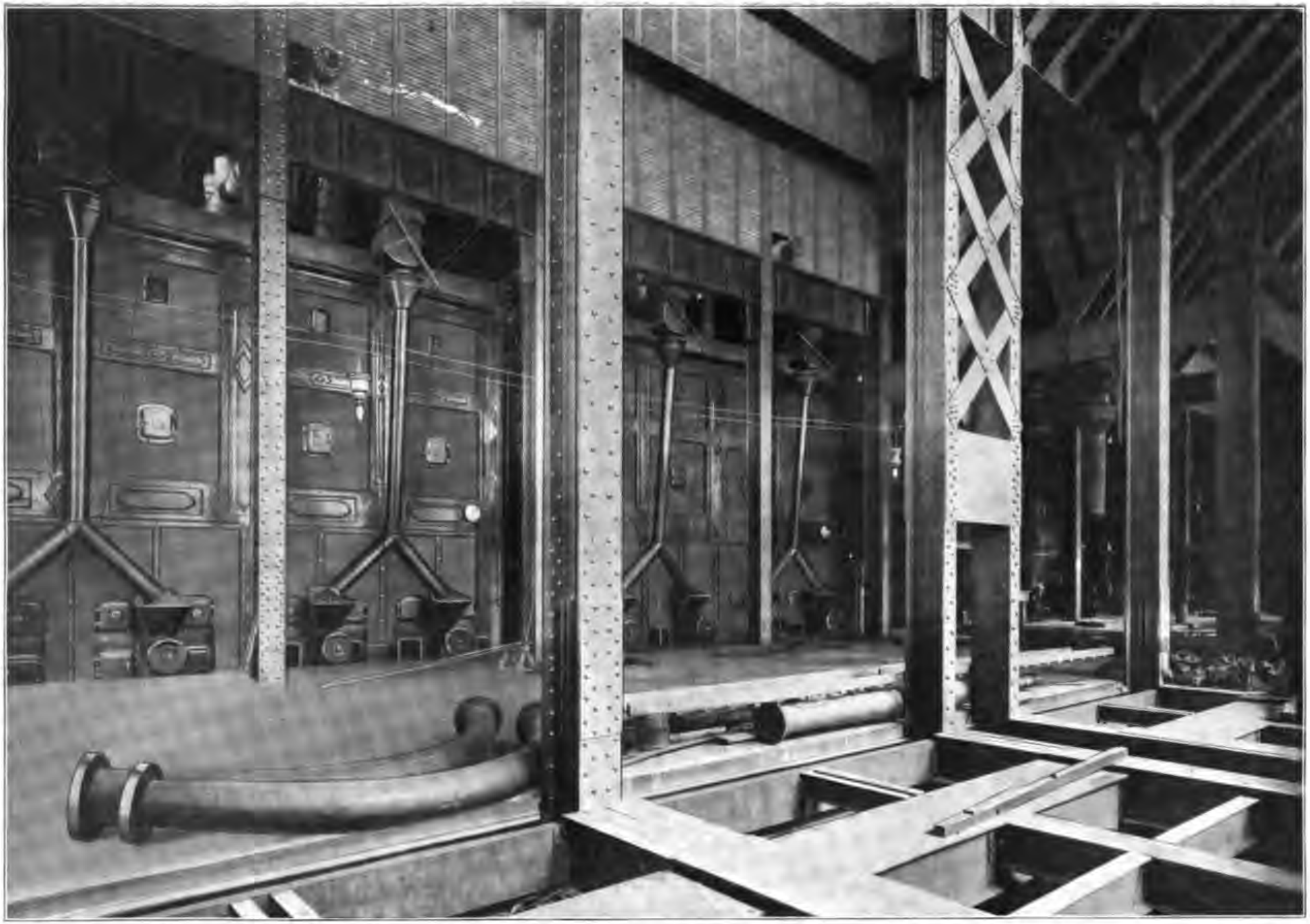
In consequence of this construction, not only may the mud drum with perfect freedom move an amount representing the resultant expansion of the boiler, but any difference in expansion between the individual tubes, such as caused when one side of the furnace is being cleaned and other side is excessively hot, is taken up by the curve in the tube. The boiler therefore stays tight, and is entirely free from the stresses and frequent leaks caused by unequal expansion of straight tubes rigidly connected at each end to headers, water-legs, or large drums.* It will thus be seen that the bent tube performs in the boiler the same function as an expansion loop in a steam line, and that its successful introduction in the Stirling boiler is a distinct and far-reaching advance in steam engineering.

Rapid Circulation—The path of the circulation in the Stirling is as follows:—The water is fed into upper rear drum, passes down the rear bank of tubes to the lower drum, thence up the front bank to forward steam drum. Here the steam formed during passage up the front bank disengages and passes through the upper row of cross tubes into the middle drum, while the *solid water* passes through the lower cross tubes into middle drum, then down the middle bank to lower drum, from which it is again drawn up the front bank to retrace its former course until it is finally evaporated. The steam generated in the rear bank passes through cross tubes to the center drum.

The temperature of gases in contact with the tubes will evidently be greatest at the bottom of the front bank, and gradually decrease as the gases proceed along their course to the breeching. Obviously then the velocity of water circulation and quantity of

FIG. 6. FORGED STEEL DRUM HEADS AND PADS FOR WATER COLUMN CONNECTIONS

*For a remarkable illustration of the effects of unequal expansion of such tubes, see photograph, pp. 546–547, in *Power*, Sept. 1904.



CINCINNATI GAS AND ELECTRIC CO., OPERATING 6,400 H. P. OF STIRIING BOILERS

steam generated will be a maximum in the front bank; in the rear bank there is a slow circulation downward equal to the quantity of water evaporated in the other two banks. The peculiar benefits arising from this action will be discussed under caption "Handling Impure Feed Water," page 19.

Rapid circulation is essential for the following reasons:

(1) To keep all parts of the boiler at practically the same temperature, thus eliminating severe stresses due to unequal expansion.

(2) To permit quick raising of steam and rapid response to sudden demands on the boiler capacity.

DRUM LUG MANHOLE AND ARCH BARS STEAM NOZZLE PAD

FIG. 7. FORGED STEEL DETAILS OF THE STIRLING BOILER

(3) To sweep away from the heating surfaces all steam bubbles as fast as formed, and thereby prevent "steam pockets" which quickly burn out the tubes. This is so particularly the case where intense local heating occurs due to use of gas or oil fuel, that some types of boiler fairly well adapted to coal cannot be successfully used with these fuels.

The third requirement is met only indifferently or not at all in those types of boilers in which tubes often numbering as many as eighteen, must discharge their entire content of steam and water through a narrow water-leg, or worse still, through a single nipple whose cross section is equal to that of but one tube. At 150 pounds gauge one cubic foot of water, when converted into steam, will have a volume of about 151 cubic feet. In consequence of this great increase in volume, as soon as the boiler is forced the nipple area becomes insufficient, steam pockets form in the lower tubes which then become overheated, and buckle and leak, and finally burn out. So inadequate are these nipples and headers that recent experiments of M. Brull have shown that in boilers whose circulation is constricted by nipples or narrow water-legs, the circulation in the upper tubes *reverses*, that is, it goes from the front to rear instead of in the opposite way as intended.* In consequence of this, much matter suspended in the water is swept into the bottom tubes, which fact, in connection with the steam pockets, explains why those tubes so rapidly fail.

In the Stirling boiler there is *no constriction of the circulation*, as each tube discharges directly into the drums, without intervention of headers, nipples or water-legs. The nearly vertical position of the tubes also promotes rapid circulation, hence steam pockets cannot form, and a fruitful cause of interrupted service and tube renewals in other types is thus eliminated from the Stirling. The record of this boiler affords incontestable evidence on this point. To cite a case: In a plant using Stirling boilers in connection with water-power, the water-wheels failed and required several days for repairing. As the service could not be interrupted the only recourse was to operate the boilers continuously at over 100 per cent. above rating until the wheels could be repaired. Considerable damage to the boiler was expected as a matter of course. When the run was over it was found that the furnace lining had been melted down † and must be renewed, but no damage of any kind to the boilers—not even

*Cf. "Appareil pour l'étude de la circulation dans les chaudières à tubes d'èau, par M. Brull," *Comptes Rendus de la Societe de l'Industrie Minerale.* Nov.-Dec., 1901. †Oil fuel was used.

9 000 H. P. OF STIRLING BOILERS, COLUMBIA CHEMICAL CO., BARBERTON, OHIO

a leaky tube—was noted, and not one penny was expended for repairing the boiler itself.

Safety—From the foregoing it is evident that the Stirling is preeminently a *safety boiler*. Since all parts are of wrought metal and either cylindrical or spherical in form, so that their strength can be accurately computed, and all flat surfaces, stay-bolts and braces have been discarded, all tendency to distortion under pressure is avoided. The design once and forever eliminates from boiler construction any necessity for that most treacherous material and fruitful source of ruptures in other types—cast iron—whether confessedly such, or disguised under such trade names as "semi-steel," "composition," "flowed-steel," "malleable metal," etc. In many countries the use of cast metal is forbidden by law.

Purchasers too often fail to realize the enormous power contained in steam and water at a high temperature under pressure, and that the energy stored in boilers is sufficient to throw them straight upward a height of from one to four miles, and they too frequently allow small differences in first cost to lead to the purchase of boilers inherently weak and dangerous.

All serious explosions result from the sudden liberation of the energy contained in large masses of steam and water. If a rupture occurs in the shell of a tubular, flue, or cylindrical boiler, the energy of all the steam and water within is suddenly released, to the destruction of the boiler itself, and frequently of its surroundings, with accompanying loss of life The same disastrous consequences attend a rupture in a water-tube boiler when the part giving away contains a large quantity of steam and water Thus, the bursting of headers in the horizontal type of water-tube boilers is frequently accompanied by the most destructive results, owing to the fact that, although they do not in themselves contain large volumes of water, they are connected with a number of tubes, which in the aggregate, contain a very large quantity of water and steam; and when the cast iron gives away, the rupture is not confined to

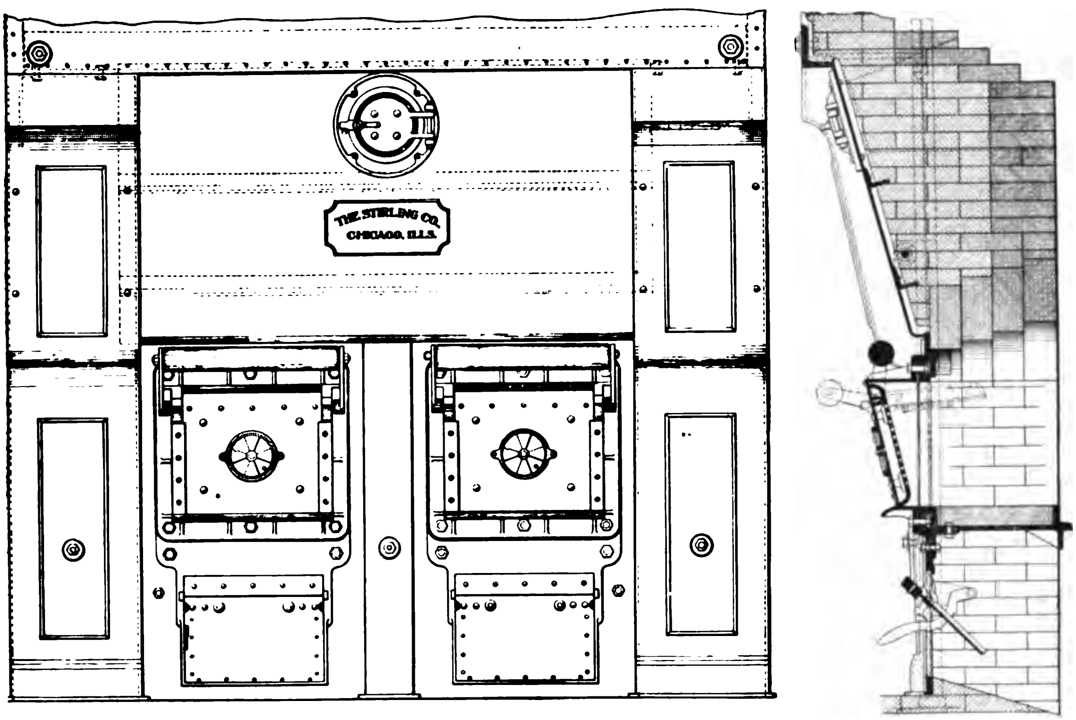
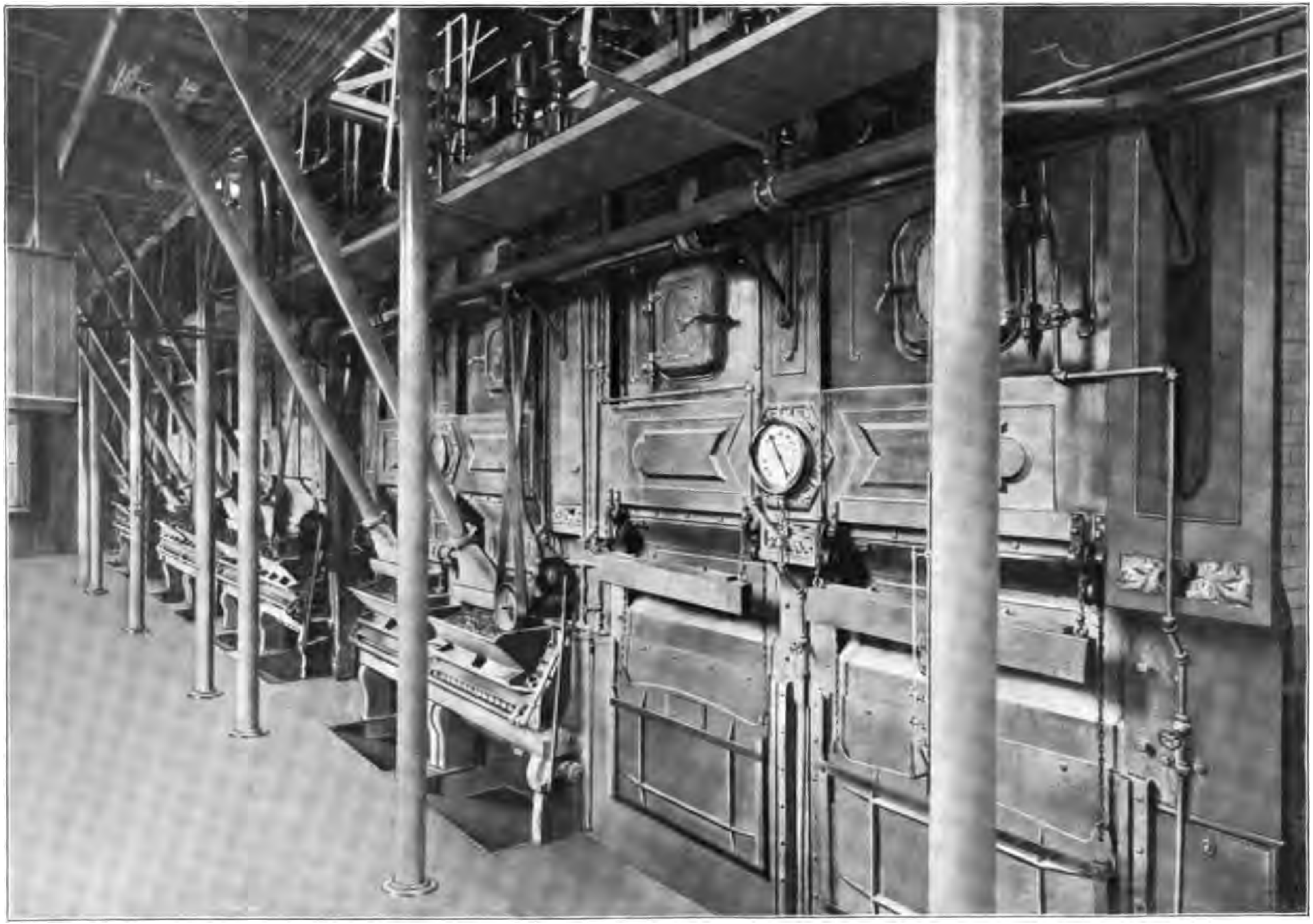


FIG. 6. STEEL FIRING AND ASH-PIT DOORS OF THE STIRLING BOILER



W. J. McCAHAN SUGAR REFINERY, PHILADELPHIA, PA., OPERATING 3,500 H. P. OF STIRLING BOILERS

a single spot, but extends throughout the entire section of the header, thus instantly liberating the water and steam contained in all the tubes expanded into it. A rupture in cast iron must, of necessity, extend throughout its entire length or breadth; while a rupture in wrought iron, or soft steel, is local, and can be enlarged only by the continued application of force. None of the so-called "safety boilers" then, in which cast metal is used, are worthy of the name. The term can be applied only to boilers in which all possible points of rupture are confined to portions such as the tubes, containing small masses of water, and which are constructed of a material in which such rupture will remain local. This condition is admirably fulfilled in the Stirling water-tube safety boiler.

Handling Impure Feed Water—In the course of the feed water from the feed drum to mud drum any precipitate formed under influence of the temperature and pressure must drop into the mud drum, which is protected from intense heat of the furnace, and acts as an excellent settling chamber. The scale-forming matter which crystallizes out under action of the temperature and pressure will deposit on the rear bank of tubes, but since the gases have passed two banks of tubes before reaching those where the deposit is formed, their temperature has been so reduced that baking of the scale to a flinty hardness is obviated, hence the deposit is soft and easily removed unless neglected for long periods of time. Even in this case the tube cannot be burned because of the low temperature of the gases surrounding it.

Hence, before passing into the front bank the water is purified,* and the danger of scale formation in the parts of the boiler that are subjected to the highest temperature is greatly reduced; consequently the interior of these tubes remains clean, and heat is transmitted more rapidly to the water, thus not only preventing the tubes from becoming overheated and burned out, but at the same time maintaining the efficiency of the boiler If there are impurities in the feed water they *must be deposited somewhere* in the boiler, hence the only recourse is to devise means to deposit them where they will do the least harm. This is accomplished by settling the precipitates where they can be blown off without gathering on the heating surface, and by causing the scale to form on the coolest heating surface where it will affect the economy the least, and remain so soft that its removal is easy. All this is accomplished to greater degree in the Stirling than in any other boiler, and in many types it is not accomplished at all. For example, in hor-

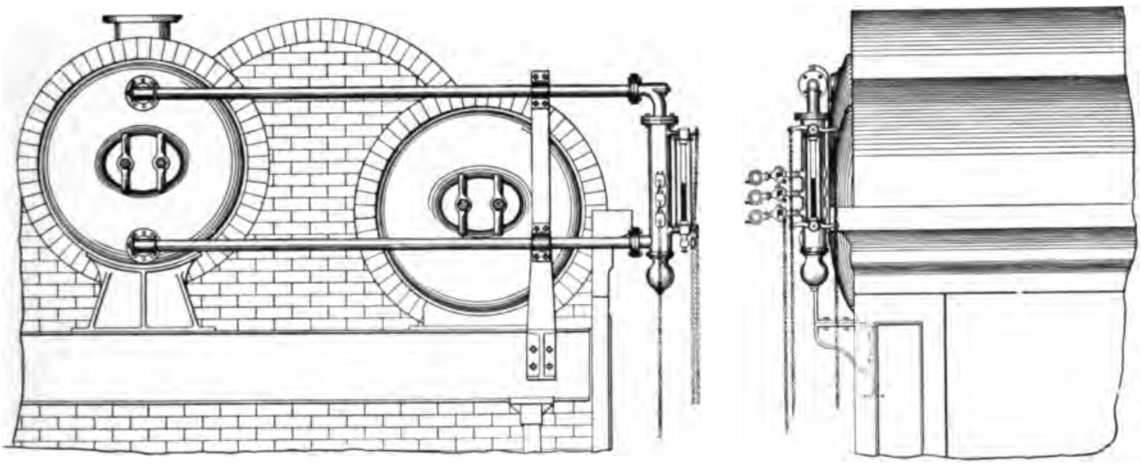


FIG. 9. WATER COLUMN AND CONNECTIONS, WITH QUICK-CLOSING GAUGE GLASS FITTINGS

*This explains why the water spaces in rear and middle drum are not connected. By compelling *all* the water to traverse the rear bank it is purified before reaching the parts of the boiler exposed to the highest temperatures.

STIRLING BOILERS AT A PLANT OF BERWIND-WHITE COAL MINING COMPANY, PHILADELPHIA, PA., 11,000 H. P. INSTALLED FOR THIS COMPANY

izontal water-tube boilers† the water is fed into upper drums, flows to the rear, then down the nipples to rear header (or water-leg), then into the tubes. Owing to the constricted area of nipple and header the velocity of water is multiplied in proportion to the number of tubes connected to one header. As the capacity of water to convey solids varies with the sixth power of the velocity,‡ only a small portion of the precipitates formed drops directly into the mud drum, while the balance is swept into the tubes. As the lower tube is the hottest, it draws in the greatest quantity of water, hence forms the greatest quantity of scale, to which is added the precipitate drawn in with the water. In consequence the deposit on the tubes is the *sum of the scale and the precipitate.* The most vital point, however, is that this deposit forms in greatest quantity in the *hottest tubes* and burns to a flinty hardness, consequently its removal is tedious and costly. Owing to the great heat on the bottom tubes, a small deposit will invariably cause the tube to burn, bag, or crack.

A most fruitful cause of burnt tubes is a piece of scale which becomes detached and falls on the bottom of the tube, and the spot under it is certain to burn out quickly The Stirling is free from this source of tube destruction, because while the scale will not form in the hotter tubes unless the boiler is neglected, even if it does form owing to such neglect and a piece becomes detached it will slide down to the mud drum instead of lodging.

Cleaning the Interior—By removing four manhole plates, which can be done in ten minutes, the entire boiler interior is accessible for cleaning. From the preceding discussion it is evident that the precipitates are settled into the mud drum, whence they are blown off at intervals; the scale is practically confined to the rear bank of tubes, and by reason of escaping the high temperatures it is soft and easily detached. Consequently it happens in most cases that only the rear bank needs cleaning each time the boiler is opened, while the others need only occasional attention. The scale is quickly and cheaply removed by a "turbine cleaner" consisting of a cutting tool driven by a water turbine attached to a hose, whose operation requires no labor beyond that necessary to guide the hose and cleaner attached, and shift it from one tube to another. Fig. 45* illustrates one of many designs of turbine cleaner on the market. So much progress has been made in the development of tube cleaners that the removal of scale from tubes (no matter whether they be straight or curved, or whether the scale be heavy or light), is merely a question of the selection of the tool or device best adapted to the work to be done. The matter is further discussed in the chapter on Boiler Cleaning, page 223.

An objection occasionally urged against curved tubes by those who have neither had experience with them nor investigated their great advantage is that they are difficult to clean, and cannot be looked through. Neither objection has weight. The turbine cleaner traverses a straight or curved tube with equal facility If it did not, it would be impossible to clean some boilers whose tubes, though originally straight, distort in service an amount often exceeding the curves in the Stirling tubes.

The thickness of scale in a tube cannot be judged by looking through the tube, because if the scale has evenly formed around the tube a difference of three-eighths of an inch in the bore cannot be detected by the eye at a point six feet away Where the incrustation is heaviest on the bottom, due to deposit of precipitate and scale, as common in all horizontal water-tubes, the departure from the round bore can be seen, and this fact doubtless lead to the belief that ability to see through a tube is an advantage. The actual fact is that the only way to *know* that there is no deposit on a tube is to pass through it a turbine tube cleaner, or a ball of proper size attached to a cord, and this test applies equally well to all tubes whether straight or curved.

Every inch of surface in the Stirling boiler can be reached and cleaned, and the time required for opening, cleaning, closing and steaming up the boiler is often considerably less than that required merely to remove

†Attempts have been made, but without success, to provide horizontal boilers with an equivalent of the Stirling mud drum. One such design is shown in *Power*, p. 565, Sept., 1904. *Page 225.

‡If the velocity is tripled the carrying capacity is 729 times as great, etc.

8,000 H. P. OF STIRLING BOILERS AT WESTERN AVENUE PLANT, CHICAGO UNION TRACTION CO.

and refit the caps over tube ends in other types of boiler.

Cleaning the Exterior—Ample cleaning doors are provided both in the sides and rear of the setting, so that the exterior of the heating surfaces may be kept clean and all accumulations of soot, ashes, etc., blown off as rapidly as they form by using a steam blower-pipe which is furnished with every boiler.*

The tubes being only slightly inclined from the vertical, there is no opportunity for soot and dust to settle on one side or on top of the tubes as in boilers of the horizontal type. Furthermore, the tubes are in parallel rows (not staggered as in some other types) and so arranged that it is possible to pass the hand, and indeed, the whole arm, between two rows of tubes, and reach those in the last tier. The exterior surfaces, then, may with ease be thoroughly cleaned of soot and kindred deposits.

Durability—By reason of the elimination of thick plates and riveted joints exposed to the fire; cast metal of all kinds; parts of irregular shape and uncertain strength; stresses due to unequal expansion; multitudes of caps, joints and nipples, and similar objectionable details, the Stirling boiler is free from parts liable to get out of order. The prevention of scale deposits in the hottest tubes; the perfect facilities for keeping the boiler clean; the rapidity of water circulation and impossibility of forming steam pockets, all combine to protect the tubes against burning out. Hence the necessity of repairs to the boiler itself is extremely remote. The setting is simple and substantial and not subject to derangement other than the natural wear of furnace lining.

In consequence the Stirling has earned an enviable reputation for durability and light cost of repairs. That this reputation is merited is evidenced by fact that according to statistics recently received from a plant operated by careful and intelligent engineers the records show that on basis of equal horse-power hours the tube renewals in the Stirling as compared with those on a prominent type of horizontal water-tube boiler

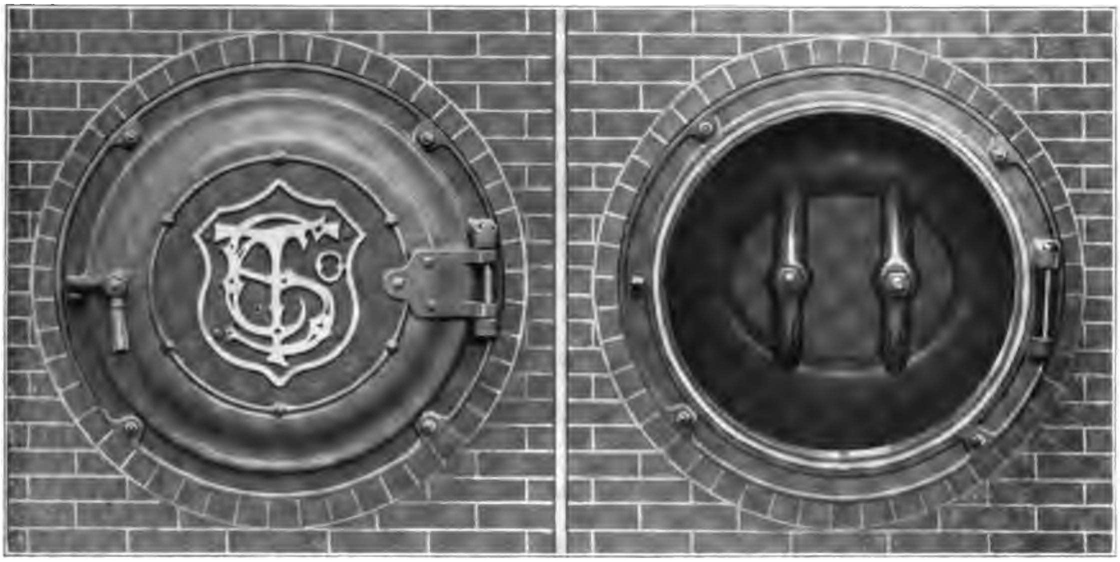
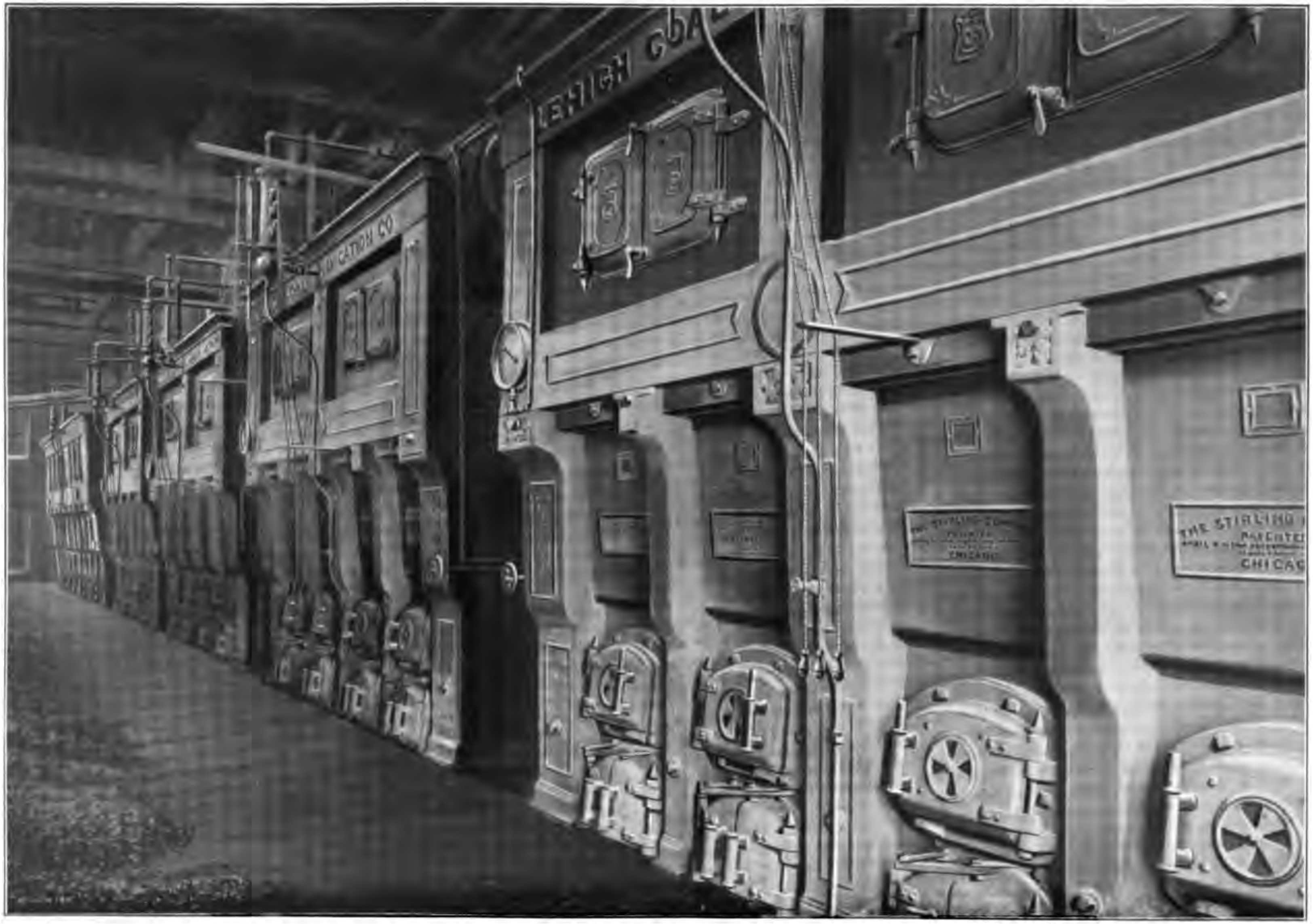


FIG. 10. IRON DOOR AND FRAME IN WALL OPENING OPPOSITE MUD DRUM

*The importance of being able to clean a boiler thoroughly, outside as well as inside ,and of keeping it clean, was well demonstrated by a comparative test made by one of our engineers some time ago on a boiler which had been allowed to run some months without cleaning, and accumulate a thickness of one-eighth inch of soot on the tubes. Before cleaning the evaporation was found to be 8.04 pounds of water per pound of coal; and after cleaning the evaporation per pound of coal was 10.30, a gain of about 28 per cent.



LEHIGH COAL & NAVIGATION CO., LANSFORD, PA., OPERATING 17,000 H. P. OF STIRLING BOILERS

of *steel header construction* were in ratio of 1 to 61, the relative maintenance account in other respects was as 1 to 3, and the labor of cleaning 6 to 35, all in favor of the Stirling.*

Facility for Making Repairs—Practically the only repairs needed to the Stirling boiler will be tube renewals, and unless the boiler is grossly neglected such renewals will be

FIG. 11. PHOTOGRAPHS SHOWING DISTENTION OF TUBES AT POINT OF RUPTURE

needed only after many years, as evidenced by fact that Stirling boilers have been in service for eight years, using hard coal, bituminous coal, natural and forced draft and oil fuel, without losing a tube or even developing a leak. Should tube renewals become necessary they are quickly and easily made.

All tube failures reduce to four classes:

(1) Pitting, which causes pin holes to be formed.

(2) Defective welds, which cause the tube to open as in A, Fig 11.

(3) An initial bagging resulting in a rupture, as in B.

(4) Scabbing and blistering as in C.

In the first case, the tube is not enlarged, and may be drawn through a tube sheet, without disturbing other tubes, though usually with difficulty owing to deposits on the outer surface.

In the other cases, the tubes become larger than their original size, hence they cannot be drawn through the tube sheet, water-leg or header, unless they are split and collapsed inch by inch for their entire length beyond the point of failure, and if they also pass through cross baffles the enlargement will pull out the bricks and destroy the baffle. To remove a tube in this way is the work of days, and in consequence the actual method used is to cut out all tubes—numbering at times half a dozen below the defective one—and to avoid destroying the baffles these tubes are cut into several pieces.

In the vertical types of water-tube boiler more recently introduced the tubes are crowded together so closely that not only is it necessary to cut out every tube in front of the defective one—numbering at times nearly a dozen—but the brickwork must be removed to gain access to the tubes

Removal of tubes from the Stirling is extremely simple. As the boiler is now constructed the tubes are spaced as in Fig. 12, and each alternate space is one-half inch wider than the tube diameter; to remove an inner tube it is merely necessary to cut the tube as near the tube-sheet as possible, pass it out through the wide space between the tubes, as indicated by the arrows in Fig. 12, and then remove it from the setting through either the side or front doors provided for that purpose. Consequently, *any tube in the Stirling boiler as now constructed may be replaced without either disturbing any other tube, distorting the tube sheet, or damaging the firetile baffles.*

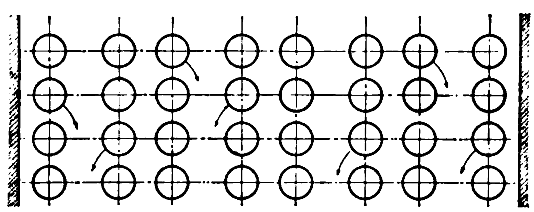
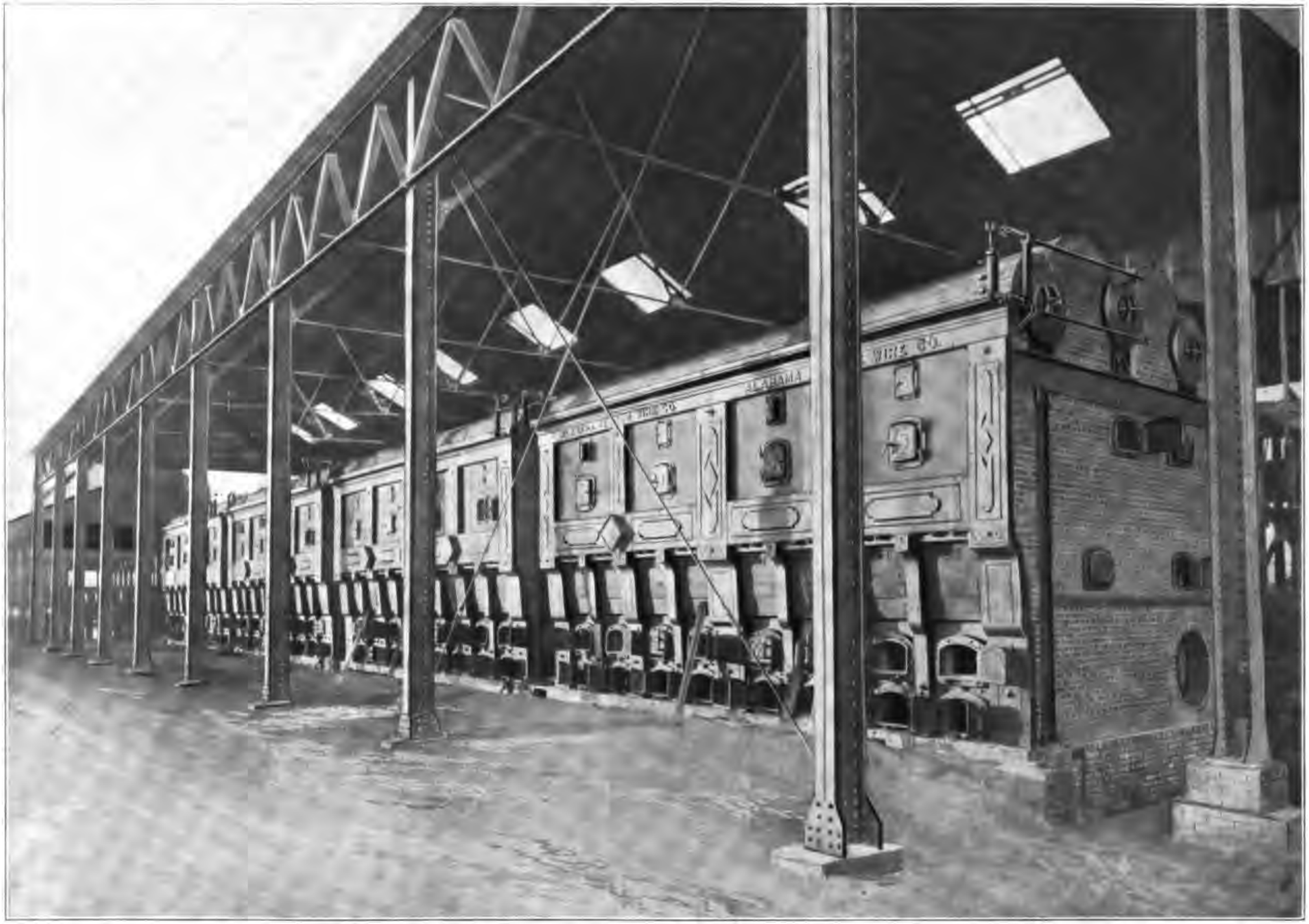


FIG. 12. TUBE SPACING IN STIRLING BOILERS

Steam and Water Space—Unless provided with sufficient steam and water space, a boiler will be subject to sudden fluctuations of pressure; the water level will be unsteady,

*Also see Table 1, page 33, referring to boilers at World's Columbian Exposition.



ALABAMA STEEL & WIRE CO., BIRMINGHAM, ALA., OPERATING 11,600 H. P. OF STIRLING BOILERS

and the steam will frequently be wet. Some prominent types of water-tube boiler use but one drum for from four to nine sections of tubes, but two drums from ten to sixteen sections, and three drums from eighteen to twenty-one sections. Hence for their entire range there are but three drum combinations, and the steam and water space varies not with the boiler horse-power but in wide jumps between combinations. In the vertical water-tube boilers there is usually for all sizes but one top drum of very limited steam and water capacity.

In the Stirling boiler there are *three upper drums*, which afford large steam and water spaces, and these vary strictly with the boiler horse-power, since increased capacity is gained, not by stacking up tubes in successive horizontal layers without increase of drum capacity, but by adding sections of tubes to the boiler width, and increasing the drum lengths in proportion

Dry Steam—The production of dry steam requires large disengaging surface; while in many types of boiler the effective disengaging surface is only a narrow strip over the nipples and water-legs (which explains why such boilers have to be provided with internal baffles of various kinds,*) in the Stirling the entire water surface of the three upper drums is available as disengaging surface, hence the steam does not disturb the water surface, and is dry. As there is no constriction of the water circulation, and as the middle drum from which the steam is drawn is somewhat higher than the other two drums, and the circulation of the water in this drum is *downward*, there is absolutely no spurting or geyser-like action of the water in this drum. The steam from the front and rear drum must also pass through hot circulating tubes which dry it before it reaches the central drum.

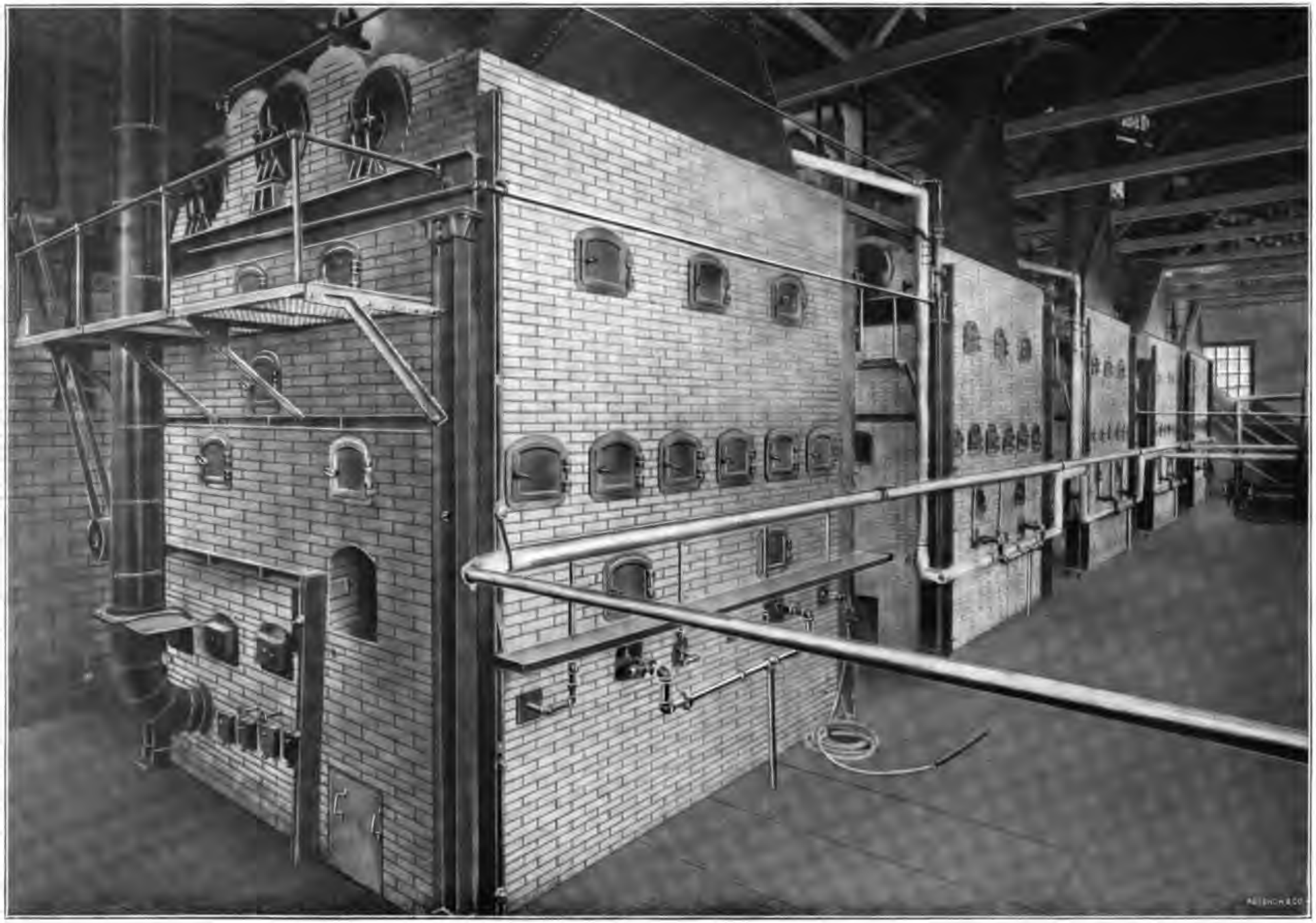
Adaptation to Different Kinds of Fuel—The large space available for furnace under the Stirling enables the grates to be proportioned for coal of the cheapest grade. By proper reduction of this grate surface, the requirements for better grades of coal can be exactly met. Should it be desired to burn wood, the most perfect form of wood furnace can be got simply by lowering the grates to level of firing floor. For burning oil or gas, the only change needed from the standard furnace is to cover the grates with fire-brick so disposed as to admit the requisite quantity of air, and to provide at rear end of the grates a loose checkerwork wall of fire-brick against which the heat will impinge. Should it be necessary to change from oil or gas to other fuel, the Stirling furnace in an hour after shutting off the burners can be made ready for firing with coal, shavings or sawdust. For burning bagasse it is necessary only to provide proper feeding apparatus and suitable grates, and the furnace thus equipped may with equal success be used for other fuels. Consequently, with but trifling changes the Stirling furnace can be adapted to *any kind of fuel*, and in no case will there be any essential departure from the general design, or removal of the arch which forms the fire-brick chamber necessary for a perfect furnace.

The Stirling furnace is also well adapted to installation of any of the various stokers in use, and to such modifications as are desirable when the boilers are installed in connection with coke ovens, heating furnaces, reverberatories for copper smelting, and other cases where the boilers are fired either wholly by waste gases, or partly by waste gases, and partly by hand.

Possibility of Driving at both Low and High Rates of Evaporation without Great Loss of Fuel Economy—This is a point of the highest importance in plants where peak loads occur. To install boiler capacity sufficient to handle the peak at regular rate of evaporation would require large initial cost, hence the usual procedure is to work the boilers above rating during the busy hours Unless the boiler can respond without material decrease in economy at the increased rate of working there will be large wastes of fuel.

The Stirling boiler meets this requirement to a degree not attainable with other types, because of its free circulation and the action of the rear bank of tubes. Here the gases come into contact with those parts of the boiler which receive the feed-water, hence the temperature difference between gases and water is a maximum, and the heat

*See "A bad case of discharge of water with steam from water-tube boilers." Vol. XXVI, *Transactions American Society of Mechanical Engineers.*



1,250 H. P. OF STIRLING BOILERS, MAHONOY PLANE PLANT, PHILADELPHIA & READING RAILROAD COMPANY. 2,250 H. P. OF STIRLING BOILERS OPERATED BY THIS COMPANY

is quickly abstracted from the gases. The economy therefore decreases very slowly as the rate of driving is increased, and this is evidenced by recent tests in which a Stirling boiler when driven at rates of 60 and 100 per cent. above rating, showed diminution in economy of but 5.11 and 7.66 per cent., respectively, below the efficiency at rating.

Adaptation to Hot-Water Heating—A unique illustration of the advantage resulting from the absence of all constriction in the path of circulation in the Stirling boiler is afforded by the extensive use of this boiler in hot-water heating plants. For such work a free passage of the water in its course through the boiler is absolutely essential. This requirement is perfectly met in the Stirling, and the same boiler, according to the necessities of the plant, is used to generate steam at one time, and at other times to heat water which is pumped through the hot-water mains.

Space Occupied—The Stirling design is so flexible that the boiler can be and is built to meet the varying requirements of height, width and depth, so that a 200 horse-power boiler can be built to occupy from 12 to 22 feet in height, 8 to 15 feet in width, and 14 to 17 feet in depth. It is therefore equally well adapted to boiler-rooms having low ceilings and ample width, as well as to those having little width and ample height. More horse-power of the Stirling type can be installed in a given number of cubic feet than of any other type on the market.

BOILER EFFICIENCY

Of all the terms relating to boiler performance, none is so much talked of, yet so imperfectly understood and erroneously applied as the word *efficiency*. It is therefore necessary that the different meanings of this word be clearly understood.

"Fuel Efficiency" is the ratio between the heat absorbed by the boiler and the heat value of the fuel burned. In nearly all cases where the term boiler efficiency occurs it is used in this sense, yet this efficiency is quite secondary in importance to another which is thus defined: "The 'Commercial Efficiency,' or the 'Efficiency of Capital' employed in the maintenance of steam generating apparatus of a given power, is measured by the ratio of quantity of steam produced to the total cost of its continuous production. This efficiency is a maximum when that cost is a minimum."* Accordingly, the Stirling boiler will be considered with respect to both of the above named efficiencies.

Fuel Efficiency—The boiler can only absorb heat, but the production of that heat depends upon the furnace, consequently the fuel efficiency is not properly boiler efficiency, but efficiency of the combination of boiler and furnace. A deficiency in either of these will affect the efficiency of the combination.

The preceding discussion has set forth the capabilities of the Stirling furnace to handle each and any kind of fuel in use, and to insure complete combustion of the gases distilled from fuels containing high percentages of volatile matter, and to prevent extinguishment of the flame by contact with cool boiler surfaces over the fire. The Stirling furnace, therefore, leaves nothing to be desired, and its efficient performance is merely a matter of proper attention from the fireman.

In regard to the Stirling boiler proper, it has already been shown: that the surfaces between the heat and water are thin, hence absorb the heat quickly; that the circulation is extremely rapid, so that the steam as fast as formed is carried away, and the heating surface kept covered with water; that the scale is formed on the coolest surfaces where it affects the economy the least, and that its removal is so easy that every inch of surface of the boiler can be kept clean and efficient; that the course of the gases in contact with the tube surface is longer than in other types of boiler, so that the heat is thoroughly abstracted; that in the rear bank of tubes the coldest water comes in where the coldest gases go out, hence the flow of water and gas is in opposite directions in conformity with Rankine's law of economy; that leaky cross-baffles have been eliminated, hence there can be no short-circuiting of the gases to the stack; that the setting is simple and tight, hence air leakages are obviated; and that there are no exposed surfaces to cause loss by condensation.

*Thurston, "*The Steam Boiler*," page 475.

SHERRY BUILDING, NEW YORK, OPERATING 775 H. P. OF STIRLING BOILERS

In consequence of these features, the Stirling boiler develops a fuel efficiency as high as ever attained under any type of boiler, and with reasonable care its efficiency will continue unimpaired with use.

In practically every case where efficiency tests are exhibited, they were made on boilers which were *thoroughly cleaned and handled by an expert.* That efficiencies thus obtained do not represent results obtainable in daily work will be evident upon considering that the moment a boiler begins a run, its surface acumulates incrustation from the water. In the preceding part of this article it has been clearly shown that in many types of boiler the deposits form on the *hottest* tubes, where beside quickly destroying the tube, they affect the economy, which rapidly falls off as the length of the boiler run increases. In consequence it will be found that after several weeks the efficiency reaches a low figure out of all proportion to the efficiency of the boiler when clean. The *effective efficiency* of the run is only the average of the efficiencies at the beginning and end of the run—a fact so seldom realized that a more general understanding of it would prove of inestimable benefit to the boiler purchaser.

In consequence of the difference between the Stirling and other types in the manner of depositing scale, it will be found that while when clean the two types of boiler may develop the same efficiency, the difference at the end of the run will be largely in favor of the Stirling. Table 60, page 208, gives results of many tests on Stirling boilers.

Efficiency of Capital Invested—Boilers are used to earn money, and what the boiler owner wants to know is "What boiler from the day I buy it until it goes to the scrap pile will return me the most money for every dollar I invest in buying and maintaining it?" Few realize that while a boiler may be efficient in fuel, it may still be a very undesirable investment. The chief factors which determine the excellence of a boiler are, in order of their importance: (1) Safety; (2) Cost of maintenance; (3) Cost of cleaning; (4) Fuel economy; (5) First cost.

Each of these has been so fully discussed in its place that the further discussion of only two of them will readily indicate the bearing of the others.

For the first case the *financial aspect of the difference in time required for cleaning the various types* will be considered. It has been shown that the time required to open, clean, close and steam up a Stirling is often less than that needed simply to remove and refit the caps in other types. The labor costs are frequently 5 to 1, and the time the boiler is off 4 to 1, both in favor of the Stirling. A point even more important, but which is frequently overlooked, is that every day a boiler is off for repairs means that much capital earning nothing, the capital being not only that invested in the boiler, but in piping connected to it, buildings housing it, and ground upon which it stands. Assume that cleaning is necessary every four weeks. In this time a Stirling will be off one day, and the cap types be off four days. The difference, three days,

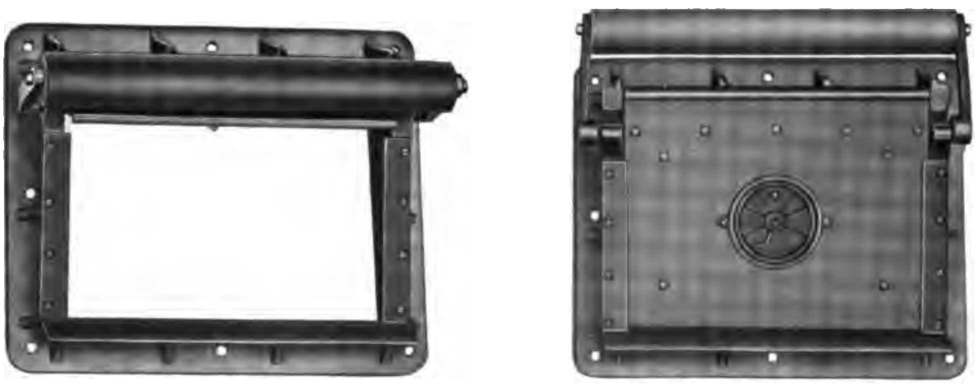


FIG. 13. COUNTERBALANCED STEEL FIRE DOORS AND FRAME

900 H. P. OF STIRLING BOILERS, PEOPLE'S RAILWAY CO., ELSMERE, DELA.

is ten per cent. of a month, hence it follows that apart from the fourfold cost of cleaning the cap type boiler, that type will per annum produce ten per cent. fewer horse-power hours, or in other words, *for the same output of steam ten per cent. greater capacity of cap type boilers than Stirlings will be necessary,* disregarding the additional time lost by the cap types, owing to more frequent tube renewals.

For the second case a comparative list of repairs of different types as installed at the World's Columbian Exposition, Chicago,

TABLE 1

A MEMORANDUM SHOWING CAUSES OF WITHDRAWAL FROM SERVICE, AND REPAIRS, ON SIX TYPES OF WATER-TUBE BOILERS, AT THE WORLD'S COLUMBIAN EXPOSITION, FROM MAY 1 TO NOVEMBER 1, 1893.

STIRLING—2700 H. P.

July	5.	Caulking Shell.
Sep.	27.	Burned Tube.
Oct.	11.	Burned Tube.

SINUOUS HEADER TYPE—3000 H. P.

May	24.	Three headers broke, one tube burst, No. 7 boiler.
June	1.	Tubes out of order.
	3.	Bad Tubes.
	8.	Bad Tubes.
	14.	Tubes leaking.
	26.	Replacing Tubes.
July	17.	Replacing Tubes.
	22.	Replacing Tubes.
Aug.	13.	Tubes out.
	22.	Changing Tubes.
	30.	Leaking Tubes.
Sep.	22.	Burned Tubes.
	25.	Burned Tubes.
	29.	Changing Tubes.
Oct.	8.	Leaking Tubes.
	10	Leaking Tubes.
	11.	Burned Tubes.
	15.	Burned Tubes.

SQUARE HEADER TYPE—3750 H. P.

July	20.	Replacing Tubes.
	25.	Replacing Tubes.
Aug.	3.	Leaking Tubes.
	7.	Leaking Tubes.
	29.	Two Tubes out.
Sep.	9.	Burned Tubes.
	12.	Leaking Tubes.
	14.	Burned Tubes.
	21.	Leaking Tubes.
	25.	Leaking Tubes.
Oct.	5.	Leaking Tubes.
	10.	Burned Tubes.
	15.	Replacing Tubes.
	26.	Engineer in charge ordered fireman not to fire No. 1 Boiler. Cause not known.

SQUARE HEADER TYPE—1500 H. P.

July	12.	Leaking Tubes.
	18.	Burned Tubes.
Aug.	5.	Burned Tubes.
	8.	Changing Tubes.
	10.	Changing Tubes.
	11.	Leaking Tubes.
	25.	Changing Tubes.
Oct.	12.	Putting in Tubes.
	15.	Replacing Tubes.
	18.	Replacing Tubes.

HEADER AND RETURN-BEND TYPE—1500 H. P.

June	28.	Replacing Tubes.
July	5.	Repairing Boiler.
	16.	Replacing Tubes.
	22.	Replacing Tubes.
Aug.	15.	Changing Tubes.
	30.	Leaking Tubes.
Sep.	8.	Leaking Tubes.
	13.	Leaking Tubes.
	15.	Changing Tubes.
	25.	Leaking Tubes.
	28.	Burned Tubes.
Oct.	8.	Burned Tubes.
	17.	Burned Tubes.
	19.	Burned Tubes.
	24.	Burned Tubes.

IRREGULARLY SHAPED HEADERS—1500 H. P.

July	5.	Repairs on Boilers.
	22.	Burned Tube.
	28.	Repairing Tube.
Aug.	1.	Repairs on Boilers.
	10.	Replacing Tubes.
	13.	Replacing Tubes.
	20.	Working on Boilers.
Oct.	17.	Leaking Tubes.

WATER-LEG TYPE 4500 H. P.

Sep.	24.	Burned Tubes.
	28.	Burned Tubes.
Oct.	4.	Burned Tubes.
	12.	Burned Tubes.
	15.	Burned T'bs in 4 B'lrs.
	22.	Burned Tubes.
	26.	Burned Tubes.

1893, will be presented, as evidence of the relative durability and repair account of the boilers when operating under identical conditions, with the same water and fuel. The purpose of this comparison being to illustrate the performance of types, and not of particular boilers, the names of the various competing boilers will be suppressed.

Imitations—One of the strongest testimonials of the excellence of the Stirling boiler is the vigor with which attempts have been and are being made to imitate it. As the circulation in the Stirling is one of its most prominent advantages, some imitators attempt to reproduce this circulation to some degree, but with an altered arrangement and number of tube banks and drums. In other respects the constructive features peculiar to the Stirling boiler are copied as closely as is thought safe.

In another class of imitations some *part* of the Stirling boiler,—as for example two drums and their connecting tubes,—is exploited as a *new type* of boiler, and great stress is laid upon the fact that *curved tubes* are used. While the curved tubes are a great advantage, these abbreviated types have merely resurrected many ancient defects which the Stirling was designed to bury. Thus, if the water is fed into their upper drum, wet steam results; if fed into the lower drum, the hottest tubes rapidly scale up, just as in the horizontal type of water-tube boilers. Besides deficient steam and water space, none of these arrangements even remotely reproduce the effect of the feed drum and rear bank of tubes in the Stirling boiler.

Conclusions—The final judgment as to the merit of a boiler must rest with those who, by long experience with it, have ascertained its virtues or its failures, and from their verdict there can be no appeal. When judged by this standard, the finding is overwhelmingly in favor of the Stirling. No other boiler ever placed upon the market has so quickly met with popular favor, retained that favor, and had such phenomenal sales. In consequence over 2,000,000 horse-power are in use, in all parts of the world where the value of human life and the economical generation of steam are understood and appreciated.

KIOTO ELECTRIC LIGHT & POWER CO., KIOTO, JAPAN, OPERATING 1,000 H. P. OF STIRLING BOILERS

Water-Tube versus Fire-Tube Boilers

In proportion as the use of steam has become more general and its economical generation has become better understood the water-tube boiler has rapidly displaced other types, until it is now used exclusively in all plants in which safety and economy are considered. Marine engineers, through excessive conservatism, have been slow in adopting the water-tube boiler, but the advantages of that type have been so clearly proved that to-day the use of water-tube boilers in the merchant marine is rapidly increasing, while the great naval powers, including the United States, have adopted for war vessels the water-tube boiler to the exclusion of other types. It must, therefore, be evident that the water-tube boiler possesses advantages which make it superior to the types it has displaced, and some of these advantages will now be set forth.

Safety—The advent of high pressure was one of the strongest factors in forcing the adoption of the water-tube boiler. To make this clear, a gauge pressure of 200 lbs., and an allowable stress of 12,000 lbs. per square inch on boiler steel will be assumed, and neglecting the weakening effect of joints, the thickness of plate necessary for cylinders of various diameters will then be,

DIA. CYLINDER, INCHES.	THICKNESS INCHES.
3½	0.020
36	0.300
48	0.400
60	0.500
72	0.600
108	0.900
120	1.000
144	1.200

The rapidity with which the plate thickness increases with the diameter is apparent; in practise all the above thicknesses, except the first, have to be augmented 30 to 40 per cent. because of riveted joints.

In water-tube boilers the drums seldom exceed 48 inches diameter, hence the thickness of plate required is never excessive. The thinner metal can be rolled of more uniform quality, the seams admit of better proportioning, and the joints can be more easily and perfectly fitted than when thicker plates are used.

The 3½ inch tube is a standard size in the Stirling boiler, and for 200 lbs. pressure a tube of No. 10 gauge is used. The thickness is 0.134 inch and with the same working stress as used in computing the above table the *safe* pressure would figure 1,072 lbs. which will indicate the margin of safety.

The essential constructive difference between the water-tube and fire-tube types is that the former is composed of parts of relatively small diameter, and a rupture of a part must of necessity be local. The drums are so disposed that they are protected from intense heat, and in the Stirling boiler there is a further advantage due to elimination of riveted joints exposed to high temperature. The greatest heat strikes on the tubes, hence the tubes are necessarily the parts which are most liable to wear and deterioration. If a tube fails it can instantly discharge only the water it contains, while the water in the other tubes must travel a considerable distance to reach the point of rupture. The quantity of water that can flow in a given time is limited by the bore of the tube, hence the results of a tube failure may cause inconvenience and require a shut down, but no considerable damage to property can be done.

Boilers of the shell type embody the undesirable necessity of "putting one's eggs all in the same basket." Not only are the shells subject to influences tending far more to rupture them than in case of drums in the water-tube type, but when they do rupture the whole body of contained water is liberated, and a disastrous and usually fatal explosion results. This is well evidenced in a recent case where a return tubular boiler made by a leading manufacturer, lately inspected and declared by competent authorities to be well constructed, and free from defects, exploded and killed 42 persons, besides causing large property loss. This typical case is merely one of a vast number

PHOTOGRAPH ILLUSTRATING THE EFFECT OF AN EXPLOSION OF A SMALL RETURN TUBULAR BOILER

which could be cited. The photograph on page 36 illustrates the disastrous results of the failure of a *small* return tubular boiler. This boiler was installed in a saw mill and exploded in November, 1904. The boiler house and the brick stack were both completely demolished, and an empty boiler adjoining the exploded one was thrown outside the building and fell beside the shed in the background. It must therefore be remembered that when boilers explode, they wreck not only themselves but contiguous buildings, hence a water-tube boiler, in addition to its other advantages, is desirable as a matter of insurance against explosion.

To the above mentioned advantages of the water-tube type the Stirling boiler adds additional advantages peculiar to itself. The elimination of all cast metal, complicated joints, riveted joints exposed to fire, stayed surfaces, and parts of irregular shape, increases the element of safety. A further advantage is the elimination of all compressive stresses. A cylinder subject to external pressure, as a fire-tube, or the internally-fired furnace of certain types of boiler, will collapse under much less pressure than it could stand if applied internally; if any initial distortion from its true shape exists, the effect of the external pressure is to increase the distortion and collapse the cylinder, while an internal pressure tends to restore it to its original shape.

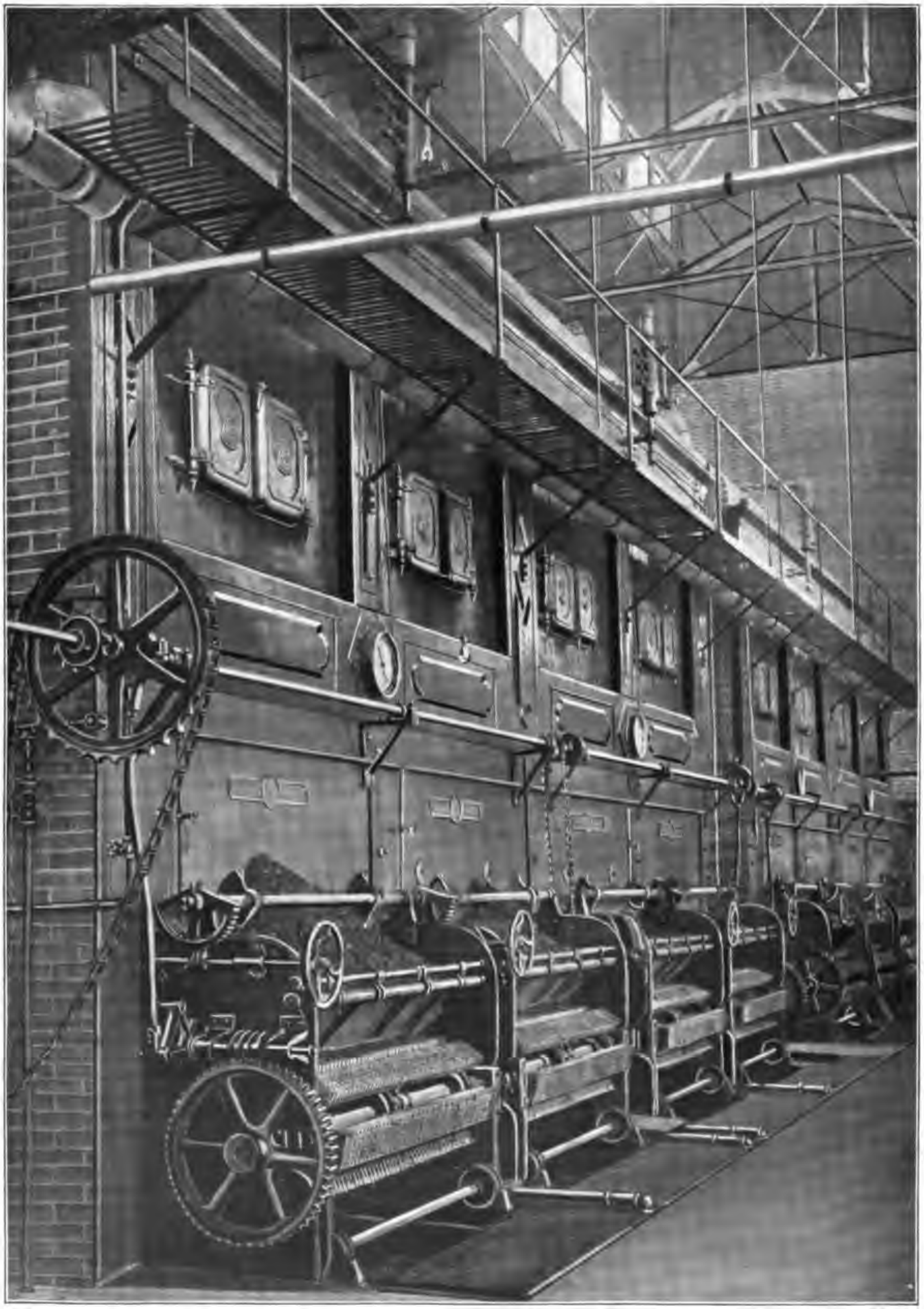
Elimination of Temperature Stresses—Stresses due to unequal expansion have been a fruitful source of trouble in fire-tube boilers, In water-legs, under internally-fired furnaces, and below the tubes, the circulation is defective. In consequence, leaks are common, and cause unsuspected corrosion in parts of the boiler that are not visible; stresses due to unequal expansion of the metal cannot be avoided, and these are often so excessive that the safety of the boiler is endangered, and many a disastrous explosion has been traced to this source.

If the temperature on the fire and water sides of a plate be kept *constant*, the rate of transmission of heat is, within reasonable limits, but little affected by the plate thickness. In practical work such constant temperatures are not maintained, owing to fluctuations due to firing, and the variation in the demand for steam. If the furnace temperature be quickly increased in response to a sudden demand for steam, the plate itself must absorb the heat to elevate its temperature, hence if the plate be thick the heat transmission to the water must be sluggish, and the steam pressure cannot be quickly increased. An even more troublesome feature in large shell boilers is the absolute necessity of firing them up very slowly, to allow their parts gradually to expand. This often takes 12 hours, and besides wasting fuel, it renders the boiler useless in emergencies. To diminish this evil artificial means of circulating the water are often used, such as "Hydrokineters" and circulating pumps, but they are merely slight palliations, and not remedies, as evidenced by the following statement from a prominent marine engineer:

"Those of us who have had to do with the maintenance of Scotch boilers know what a continual round of expensive repairs have to be made at nearly every inspection, and in almost every case the causes are due to straining because of unequal expansion. * * * There are Scotch boilers running at a working pressure of 150 lbs., upon the bottom of the shell of which the bare hand may be placed without any inconvenience."

These troubles are wholly obviated in the Stirling water-tube boiler. The metal exposed to heat is thin, hence the pressure rapidly responds to an increase in the furnace temperature. Circulation is rapid and takes place in a *definite path* which is arranged in conformity with the law of greatest economy. The rapid circulation practically equalizes the temperature in all parts of the boiler, and the arrangement of parts is such that temperature stresses are eliminated. Leaks and corrosion due to them are obviated, and the repair bill is lessened.

Quick Steaming—The thin metal in the tubes and the elimination of temperature stresses in the Stirling boiler permit steam to be raised so rapidly that in emergencies the boiler can be pressed into service and operated at a high capacity, long before a boiler of the shell type could safely be brought up to pressure. A unique illustration of



FORD PLATE GLASS CO., TOLEDO, O., OPERATING 4,000 H. P. OF STIRLING BOILERS

the adaptability of the water-tube boiler in situations where sudden loads are to be encountered is afforded by a plant generating current for electric locomotives pulling trains through a long tunnel. When no train is passing there is no load on the plant, the engines turn slowly, and the boilers have little to do. A few moments before a train arrives a signal is given, the draft is turned on the boilers, steaming at full rate at once begins, the engines are speeded up, and the train upon arrival at the tunnel is at once pulled through. The method of operating the water-tube boilers saves a large amount of fuel which would be necessary for any other type of boiler which cannot almost immediately respond to sudden demands for steam.

Cleaning—In order that a boiler may be cleaned thoroughly it is necessary that every inch of its interior surface be accessible. This requirement cannot be met in fire-tube boilers. The tubes are nested together, and when incrustation forms upon them it can be removed only from such surfaces as can be reached. With a space between tubes often less than 1¼ inches all that can be done is to pass in the vertical spaces a thin sharp-pointed tool which can remove only a limited amount of the deposit on the side of the tube. In consequence nearly the entire tube circumference is inaccessible. The efficiency of the boiler rapidly falls off, and if the tubes get very hot they burn, so that frequent renewals are necessary. In the Scotch marine type, even when the engines operate condensing, tube renewals at intervals of six to seven years are necessary, and renewals in less than a year are sometimes required. In return tubulars operated with very bad water annual tube renewals are not uncommon. In the return tubular much sediment falls on the bottom sheets where owing to the great heat it bakes to such excessive hardness that the only method of removing it is to chisel it out. This can be done only when sufficient tubes are omitted to leave space for a man to crawl in, and the discomforts under which he must work are apparent. Unless this deposit be removed, a burned and bagged plate will be the inevitable result, and unless attended to in time an explosion will follow.

The deposit of mud in water-legs of some types of boiler is an active agent in causing corrosion, and the difficulty of removing this deposit through hand holes is well known. A complete removal is practically impossible, and as a last resort to obviate corrosion it is common to make the bottom of the water-legs of copper.

The soot and fine coal swept along by the draft will settle in the fire-tubes, and unless promptly removed it often hardens so that it must be cut out with a special form of scraper. It is not at all unusual, when soft coal is used, to find the fire-tubes half filled with soot, which not only renders useless a large part of the heating surface, but diminishes the draft, so that it is difficult to develop the heat necessary to secure capacity from the heating surface that is left.

The effects above named are diminished in varying degrees in some water-tube boilers, but are wholly obviated in the Stirling. The manner in which this boiler handles impure water and minimizes formation of scale has been described on page 19, and the methods of removing this scale are given in the chapter on Boiler Cleaning. Every inch of interior surface can be reached and *kept clean.*

The deposit of soot on the outside of a horizontal tube is less than when deposited inside, but it is nevertheless sufficient greatly to reduce the effective heating surface. The nearly vertical tubes in the Stirling obviate this, since none of the fine material carried over by the draft can rest on the tubes, and the only deposit will be the soot or tarry matter which condenses from the gases. Even this can be blown off while the boiler is under pressure, while to expose the tube sheet of a fire-tube boiler when under steam would be a hazardous risk, because of the sudden contraction due to inrush of cold air.

Efficiency—What a boiler *may* do when clean, and what it *does* do when foul are very different things, and the magnitude of the difference is seldom understood by owners of boilers. It must be remembered that the moment a boiler begins work its heating surface begins to foul both inside and out. When a boiler has been operated several months without cleaning its efficiency may drop off by as much as 30 per cent. or more,

ARMOUR INSTITUTE, CHICAGO, ILL., OPERATING 1,150 H. P. OF STIRLING BOILERS

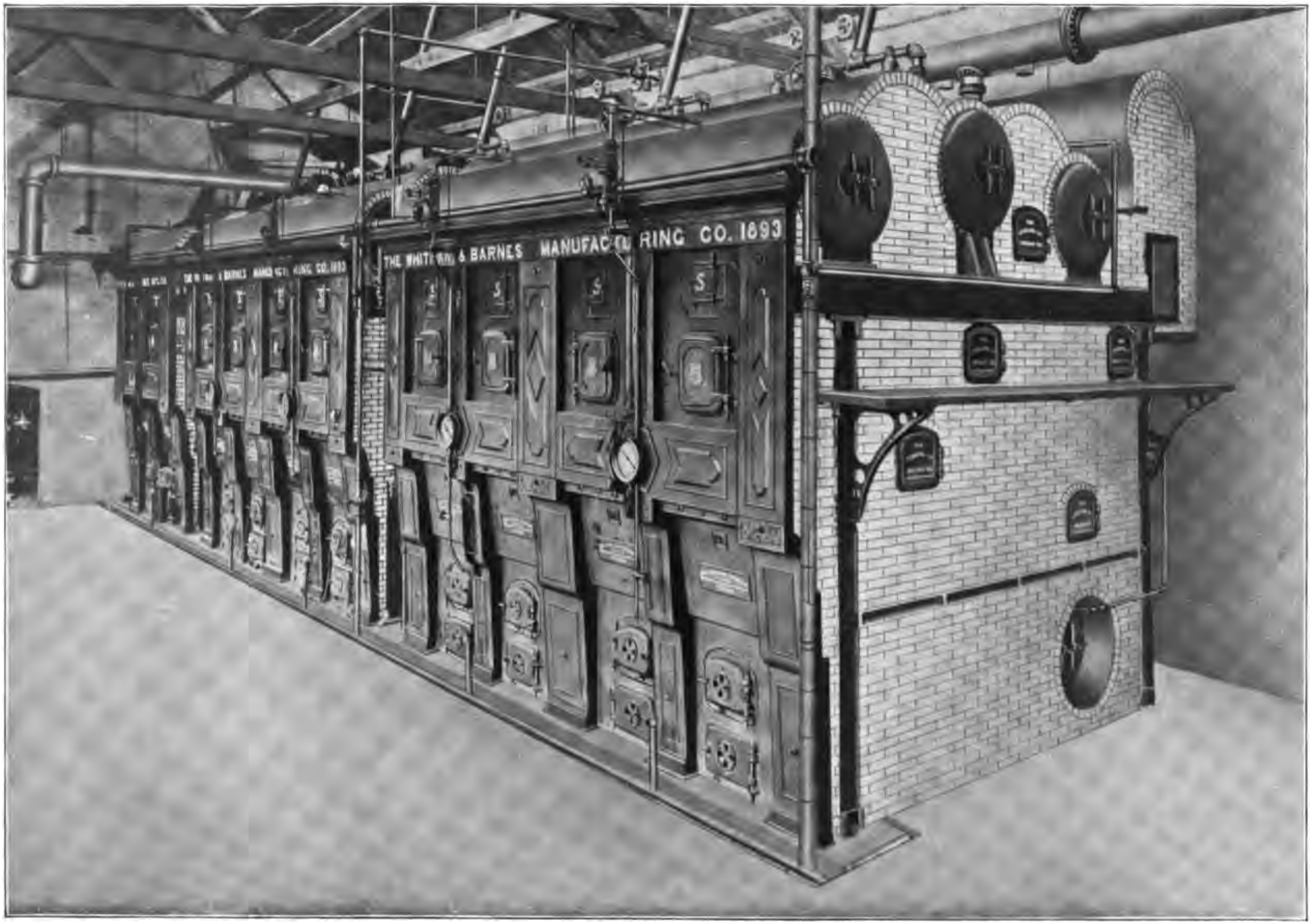
and in case of very bad water this result may happen in a much shorter time. The results will be worse in proportion as the deposits form on the hotter surfaces, yet in the return tubular type the sediment drops to the bottom of the shell, sweeps forward where the surfaces are hottest, and drops where it affects the economy the most. The tubes then foul up, and the impossibility of thoroughly cleaning them has been pointed out. The readiness with which deposit forms on crown sheets of fire-box boilers, and the large furnaces of internally-fired types is too well known to require comment.

In the Stirling boiler the action is entirely different. The sediment is removed and blown out, and the scale forms on the coolest part of the boiler, as explained on page 19, hence the initial efficiency is not only higher than in the other types because of the superiority of design, but it does not diminish so rapidly when the boiler is in use.

Efficiency depends not only upon the degree to which the boiler absorbs heat, but also on the degree to which the furnace can develop the heat to be absorbed. The poorer the fuel the more impossible it is to develop its heating value when the gases, before the combustion is completed, come into contact with the shell as is the case in return tubulars, or with furnace walls surrounded by water, as in internally-fired types. The furnace must be practically enclosed in fire-brick, and this requirement is perfectly met in the Stirling furnace. The matter is further discussed in the chapters on "Fuel Burning" and "Steam Boiler Efficiency."

Repairs—The possession of great strength; the elimination of stresses due to uneven temperatures, and of leaks and the corrosion due to them; the protection of the drums from external heat; and prevention of deposits on the hottest tube surfaces, all unite to obviate necessity of repairs. The tubes are the only parts in the Stirling boiler which may need renewal, and then only at infrequent intervals. Such renewals can be quickly and cheaply made. In fire-tube boilers tube renewals are a much more serious undertaking. As the tubes are enlarged by accumulations of hard deposit they cannot be drawn through the tube sheet unless they are collapsed inch by inch for their entire length; consequently it is usual to cut out all tubes necessary to give access to the defective one, and the tubes so cut are passed out of the manhole. In case of a bagged or blistered sheet the defective part must be cut out by hand, tap holes be drilled by ratchets, and as it is impossible to get space in which to drive rivets, a "soft patch" is necessary. This is only the sorriest of makeshifts, and usually will result in requiring the working pressure to be reduced, or a new plate to be put n. To do the latter the old plate must be cut out, a new one must be scribed to place so as to locate rivet holes, and in order to secure room in which to work when driving rivets the boiler must be retubed. The setting of course must be partly torn down, and then replaced, so that the final cost wil usually be considered greater than the initial cost of a Stirling boiler. In case of a rupture the water-tube boiler would lose a tube or two which can be quickly replaced; the fire-tube boiler will be so completely demolished that the question of repairs will be shifted from the boiler to the surrounding property, and the damage done to this property will usually exceed many times the cost of a boiler of a type which would have eliminated all possibility of the explosion. The boiler purchaser must consider that not only are the *current repairs* of the Stirling much less than required for the fire-tube types, but that as a business proposition it is not wise to invest large sums in equipment which, through a possible accident to the boiler, may be either wholly destroyed, or so damaged that the cost of repairing it, and the loss of business until the repairs are made, would purchase boilers of absolute safety and leave a large margin beside. Add to this the possible loss of human life, and *the true repair account* to be considered when purchasing a boiler will receive more consideration than is usually accorded to it.

Adaptability—The superority of the water-tube type when sudden loads are frequent has been pointed out. It is often contended that the fire-tube boiler is preferable to the water-tube when operating under variable loads, for the alleged reason that the greater amount of water in the shell type acts as a reservoir of heat, so that upon a reduction



WHITMAN & BARNES MFG. CO., WEST PULLMAN, ILL., OPERATING 2,300 H. P. OF STIRLING BOILERS

in the steam pressure the stored heat immediately generates sufficient steam to meet the demand. In reply it need only be said that so far as the Stirling is concerned it often contains per square foot of heating surface, or per horse-power, as much water as the return tubular, and much more than some other types. Apart from this, the argument is also unsound. The total heat of steam at 150 lbs. gauge pressure is 1193.5 B. T. U., and at 100 lbs., 1184.9 B. T. U.; difference 8.9 B. T. U. As the latent heat of steam at 100 lbs. gauge is 876.5 B. T. U. it will be seen that a drop of 50 pounds would be necessary to provide heat enough to evaporate only one per cent. of the water in the boiler, consequently a drop of sufficient magnitude to have any practical influence in generating extra steam would go beyond the limits which any engineer would tolerate. The locomotive boiler, which is subjected to violent fluctuations of load often contains not over *one-third* as much water as a Stirling boiler developing the same power

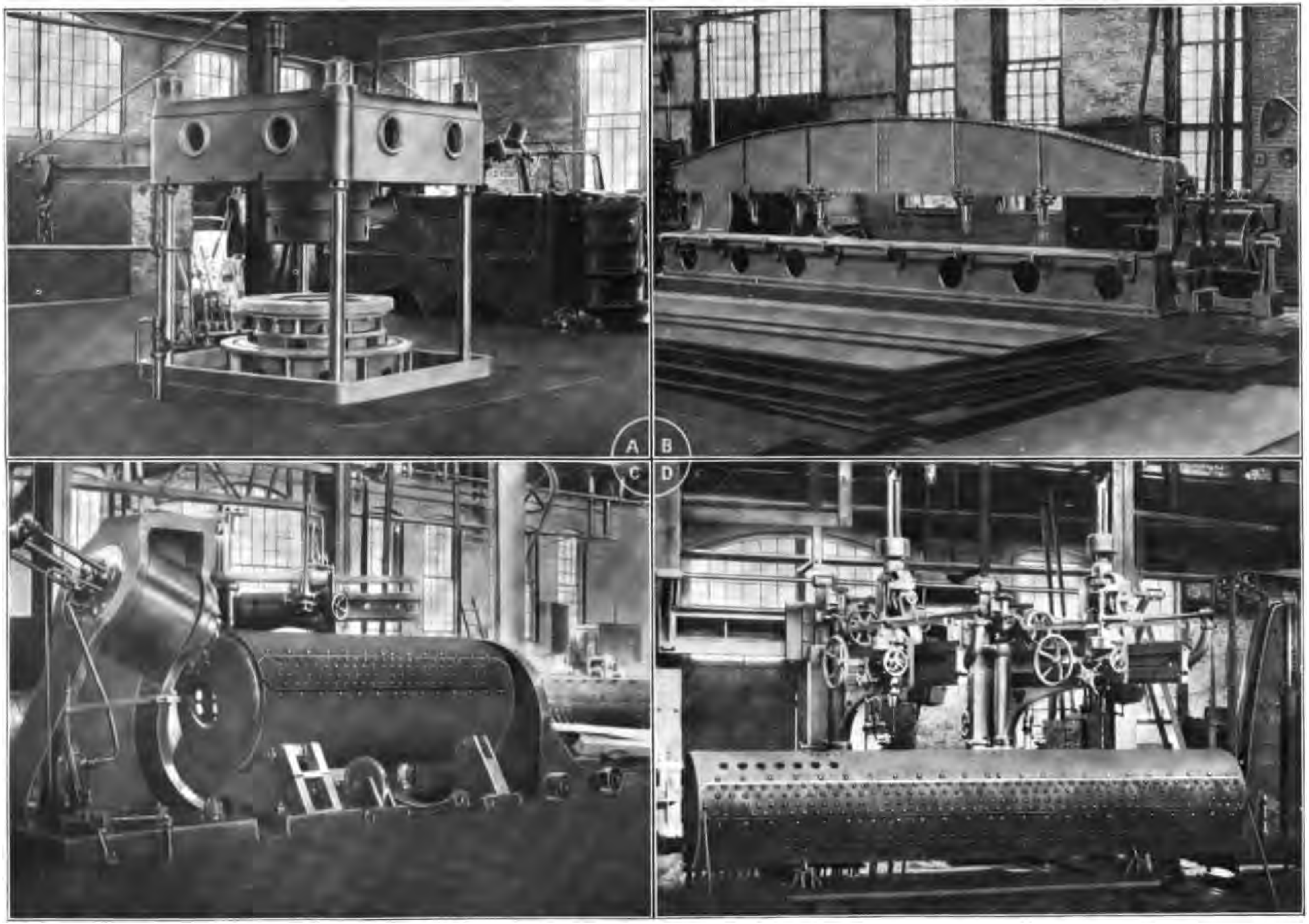
A defect of all shell types of boiler is that once they are built there can be no adjustment of draft areas to suit either the chimney to which the boilers are attached or the fuel which is to be burned. Many water-tube boilers are equally faulty in this respect. In the Stirling boiler it is possible to adjust the draft area to suit any conditions by shortening or lengthening the firetile baffles. To do this it is necessary merely to take out or to add on a few tiles, without changing bridge walls, or flame plates, or other parts difficult of access or expensive to alter. Consequently it is the work of only a few hours to adjust the draft areas in the Stirling to suit any new fuel which may have to be used, while such adjustment cannot be made at all in any type of fire-tube boiler.

Space—The cost of a boiler plant must include not only the boilers but the ground, the buildings, piping, stacks, breechings, coal bins, and everything else required to complete the plant ready to run. Obviously a saving in space occupied by the boiler will effect a saving in piping and buildings. The Stirling boiler occupies so much less space than that required by fire-tube boilers of the same capacity that the saving thus made possible will often amount to considerable percentage of the cost of the boilers. For example, 175 H. P. is about the limit of size of the return tubular; a boiler of that capacity would be 78 inches diameter and 18 ft. long, and when erected the setting would require a space 23 ft. long and 9 ft. 6 inches wide. A Stirling boiler of the same capacity can be installed in a space 16 ft. 3 inches long by 10 ft. wide, 18 ft. 10 inches long by 7 ft. wide, or in other lengths and widths to conform to the requirements.

In large installations the showing is still more marked in favor of the Stirling: thus three of the above tubulars would require a space 26 feet 6 inches wide by 23 feet deep. The equivalent 525 H. P. of Stirlings would require a space 23 by 16 feet, or 17 by 19 feet, or intermediate widths and depths. Similarly, six of these tubulars would require a space 52 by 23 feet, while equivalent Stirling boilers could be placed in a space varying from 35 by 17 feet to 29 by 19 feet. The additional aisle space at the ends and rear would be the same for both. Because of the greater space required by shell boilers an attempt is often made to increase their heating surface by crowding the tubes very close. The effect of this is to increase the difficulty of removing scale from the tubes, and to cause excessive moisture in the steam.

Should the growth of the plant require the substitution of larger boilers the water-tube type can be taken apart, removed and replaced by larger units, without alteration of buildings, and installation of expensive tackle. This is an item of great importance when boilers are installed under buildings, since fire-tube boilers installed in such places usually cannot be taken out, without either cutting them to pieces or tearing down parts of the building.

Other important advantages of the water-tube over the fire-tube boilers will be pointed out in the chapters which follow.



VIEWS FROM DRUM SHOP OF THE STIRLING COMPANY'S WORKS

A. HYDRAULIC HEAD-FLANGING MACHINE. B. A PLATE PLANER. C. HYDRAULIC PRESS FOR INSERTING DRUM HEADS. D. DOUBLE ARM RADIAL DRILL

Works of The Stirling Company

The Stirling Company manufactures Water-Tube Safety Boilers for both stationary and marine use, superheaters, chain grates, bagasse furnaces and conveyors, stacks, breechings, etc. Its works are located at Barberton, Ohio, and occupy 60 acres; the shop floor space under roof is 300,000 square feet; there are 24 separate shops, and 28 other buildings. The shops and offices are constructed of steel, brick, slate, and wood, on concrete foundations, and reflect the highest development in American factory construction. Fire protection is afforded by the automatic sprinkler system, and fire hydrants located throughout the grounds. These connect with the Company's private system of fire mains and pumps, and the City Service can be used in addition if necessary.

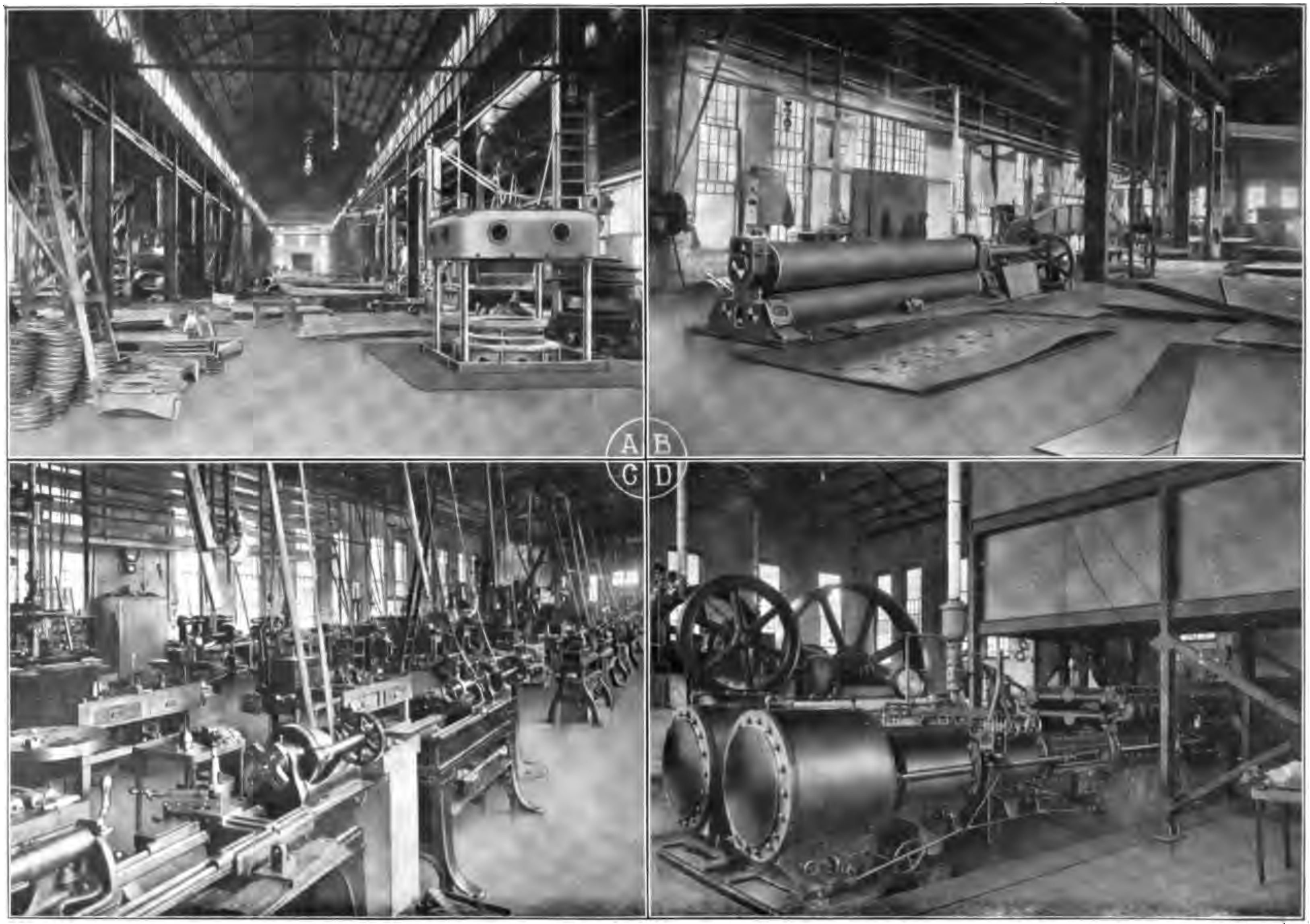
The Works were started in 1890, since which time their valuation and capacity have increased over tenfold, yet the steady growth of the Company's business is such that constant additions to the equipment are demanded. Already the plant is the largest in the world which is devoted exclusively to manufacture of water-tube boilers. The boilers supplied by the Company are manufactured in its own Works, under the superintendence of its own engineers. All material is rigidly tested, and every precaution that years of experience can suggest is taken to insure that both the material and the workmanship of these boilers are of the highest grade obtainable.

All parts of the plant are provided with standard gauge railway tracks and switches, and the Company owns an excellent equipment of locomotive cranes, cars, and buggies adapted to the special service to be performed. A unique feature of the works is an immense gantry crane with double cantilever arms spanning the entire drum yard. Another overhead crane operates through a distance of 600 feet from the foundry to the fitting shop, and passes directly into both buildings.

The Works are equipped with steam, electric, hydraulic and pneumatic power supplied from a central station, and all buildings are electric lighted. The shop tools are for the most part electrically driven, while hydraulic power is used for riveting and flanging, and pneumatic power for caulking, etc.

Many of the tools have been designed by the Company and embody every known improvement for accomplishing the result for which they are intended. The entire equipment is a striking example of modern American practise in economizing labor and material, and the Company is ever on the alert to adopt any method or improvement which may reduce cost of manufacture or increase the excellence of its product.

In addition to the Stirling boiler as described in this catalog, the Company is extensively engaged in the manufacture of water-tube boilers specially designed for marine use, and has already installed this type of boiler in the Russian cruiser Variag, the Russian battleship Retvizan, the United States battleships Maine, Virginia and Georgia, the cruisers Colorado and Pennsylvania, the monitor Nevada, the Imperial Ottoman cruiser Medjidia, a private steam yacht, and several of the largest vessels in use in the merchant marine. Work is now actively progressing on boilers for several steamships under construction for service on the Great Lakes.



VIEWS FROM SHOPS OF THE STIRLING COMPANY, BARBERTON, OHIO

A. VIEW IN DRUM SHOP. B. ONE OF THE PLATE ROLLS C. MACHINE SHOP VIEW. D. PART OF POWER PLANT

Heat

Heat is a condition of matter caused by vibratory motion among its particles. Very hot bodies are those in which the vibrations are very rapid, and the hotter the body, the more rapid the vibrations.

Temperature—The temperature of a body is the measure of its capability of communicating to adjacent bodies sensible heat, or heat that may be felt. When two bodies of different temperature are placed into contact the hotter body becomes cooler, and the colder body hotter, until finally their temperatures equalize. This proves that heat can be transferred.

Heat Effects—When heat is added to or taken from a body, either the temperature of the body is altered, or its volume is varied, or its state is changed. Thus, if heat be added to water under atmospheric pressure, the temperature of the water increases until it reaches 212° F. If more heat be added and the pressure remains unchanged, the temperature does not further increase, but the water evaporates into steam. Heat thus changes water from a liquid to a gaseous state. If heat be abstracted from water the temperature is reduced until it reaches 32° F., after which any diminution of heat does not further decrease the temperature, until the liquid is converted into a solid, or ice. The quantity of heat passing from one body to another can thus be estimated by the effects produced. Therefore heat is something that can be both transferred and measured.

The general effect of heat on a body is to increase its volume. If heat be abstracted from a body the contrary effect ensues, and the volume is diminished. Hence the general principle, to which, however, there are some exceptions, that heat expands and cold contracts. These effects, arising from a change of temperature, are produced in very different degrees according to the nature of the bodies. They are small in solids, greater in liquids, and greater still in gases.

It is well known that the work expended in friction apparently is lost as regards mechanical work; that heat is developed when friction occurs; that the greater the friction the greater is the amount of heat produced. Experiments have proved that the amount of heat generated by friction is exactly equivalent to the amount of work lost, whence it is shown that heat like mechanical work is one of the forms of energy.

Thermometers—In consequence of the uniform expansion of mercury and its great sensitiveness to heat, it is the fluid most commonly used in the construction of thermometers. In all thermometers the freezing and the boiling point of water, under mean atmospheric pressure at sea level,

TABLE 2

COMPARISON OF THERMOMETER SCALES

	Fahrenheit	Centigrade	Reaumur
Absolute Zero . . .	−460.66	−273.70	−218.96
	0	−17.77	−14.22
	10	−12.23	−9.77
	20	−6.67	−5.33
	30	−1.11	−0.88
Freezing Point . .	32	0.	0.
Maximum Density of Water . . .	39.1	3.94	3.15
	50	10.	8.
	75	23.89	19.11
	100	37.78	30.22
	200	93.34	74.66
Boiling Point . . .	212	100.	80.
	250	121.11	96.88
	300	148.89	119.11
	350	176.67	141.33

$$F = \tfrac{9}{5} C + 32° = \tfrac{9}{4} R + 32°$$
$$C = \tfrac{5}{9} (F - 32°) = \tfrac{5}{4} R.$$
$$R = \tfrac{4}{5} C = \tfrac{4}{9} (F - 32°).$$

are assumed as two fixed points, but the division of the scale between these two points varies in different countries, hence there are in use three thermometers, known as the Fahrenheit, the Centigrade or Celsius, and the Reaumur. In the Fahrenheit the space between the two fixed points is divided into 180 parts; the boiling point is marked 212, and the freezing point is marked 32,

and zero is a temperature which, at the time this thermometer was invented, was incorrectly imagined to be the lowest temperature attainable. In the Centigrade and the Reaumur scales the distance between the two fixed points is divided into 100 and 80 parts, respectively. In each of these two scales the freezing point is marked 0, and the boiling point is marked 100 in the Centigrade, and 80 in the Reaumur. Each of the 180, 100, or 80 divisions in the respective thermometers is called a degree.

Table 2 and appended formulas are useful for converting from one scale to another:

Absolute Zero—Experiments show that at 32° F. a perfect gas expands $\frac{1}{492.66}$ part of its volume if its pressure remains constant, and its temperature is increased one degree. If this rate of expansion per degree held good at all temperatures (and experiment shows that it does above the freezing point), the gas, if its pressure remained the same, would double its volume if raised to a temperature of $32 + 492.66 = 524.66$ Fah., while under a diminution of temperature it would shrink and finally disappear at temperature of $492.66 - 32 = 460.66°$ below zero Fah. While undoubtedly some change in the law would take place before the lower temperatures could be reached, this is no reason why the law may not be used within the range of temperatures where it is known to hold good. From the preceding explanation it is evident that, under a constant pressure, the volume of a gas will vary as the number of degrees between its temperature, and the temperature of —460.66° Fah. To simplify the application of the law, a new thermometric scale is constructed as follows: the point corresponding to —461° F. is taken as the zero point on the new scale, and the degrees are identical in magnitude with those on the Fahrenheit scale. Temperatures referred to this new scale are called *absolute temperatures*, and the point —461° F. (= −273 C) is called the *absolute zero*. To convert any temperature Fahrenheit to absolute temperature, add 461 to the temperature on the Fahrenheit scale; thus 54° F. will be $54 + 461 = 515°$ absolute temperature; 113° F. will likewise be equal to $113 + 461 = 574°$ absolute temperature. If one pound of gas had a temperature of 54° F., and another pound had a temperature of 113° F., the respective volumes would be in the ratio of 515 to 574, if the pressure on each were the same.

British Thermal Unit—The quantitative measure of heat is the British Thermal Unit. It is ordinarily written B. T. U., and is the quantity of heat required to raise the temperature of a pound of pure water one degree, at its point of maximum density, viz.: 39.1° F. In the metric system the unit is the *calorie*, or the heat necessary to raise the temperature of a *kilogramme* of water one degree Centigrade, at the point of maximum density.

1 B. T. U. = .252 Calorie

1 Caloric = 3.97 B. T. U.

Specific Heat—The quantity of heat required to raise the temperature of unit weight of any substance one degree varies with the substance, and is called the specific heat of that substance. It is also the ratio of the heat so required to that required to heat the same weight of water. For solids, at ordinary temperatures, the specific heat is constant for each individual substance, although it is variable at high temperatures. In the case of gases a distinction must be made between specific heat at constant volume, and a constant pressure.

Where merely specific heat is stated it implies specific heat at ordinary temperature, and *mean* specific heat refers to the average value of this quantity between the temperatures named.

The specific heat of a mixture of gases is obtained by multiplying the specific heat of each constituent gas by the percentage of that gas in the mixture, and dividing the sum of the products by 100. The specific heat of a gas whose composition is CO, 13; CO_2, 0.4; O, 8; N, 78.6. is found as follows:

$$
\begin{array}{llcl}
CO, & 13 & \times .217 = & 2.821 \\
CO_2, & 0.4 & \times .2479 = & .09916 \\
O, & 8 & \times .21751 = & 1.74008 \\
N, & \underline{78.6} & \times .2433 = & \underline{19.16268} \\
 & 100.0 & & 23.82292
\end{array}
$$

and $23.8229 \div 100 = .238$ = specific heat of this gas.

Latent Heat—Where the application of heat results in a change of state of a substance, either from solid to liquid, or from

liquid to gaseous, there is an absorption of heat without any rise in temperature, and the heat thus absorbed is termed *latent* (or hidden), because it apparently disappears, and is not measurable by a thermometer.

It is not lost, but reappears whenever the substance passes through the reverse cycle, from a gaseous to liquid, or from a liquid to a solid state. Latent heat is therefore the quantity of heat which apparently disappears, or is lost to thermometric measurement, when the molecular constitution of body is being changed. It is expended in performing the work of overcoming the molecular cohesion of the particles of the substance, and in overcoming the resistance of external pressure to change of volume of the heated body.

TABLE 3

SPECIFIC HEATS

SOLIDS.	
Copper	.0951
Gold	.0324
Wrought Iron	.1138
Cast Iron	.1298
Steel (soft)	.1165
Steel (hard)	.1175
Zinc	.0956
Brass	.0939
Glass	.1937
Lead	.0314
Platinum	.0324
Silver	.0570
Tin	.0562
Ice	.5040
Sulphur	.2026
Charcoal	.2410

LIQUIDS.	
Water	1.0000
Alcohol	.7000
Mercury	.0333
Benzine	.4500
Glycerine	.5550
Lead (melted)	.0402
Sulphur (melted)	.2340
Tin (melted)	.0637
Sulphuric Acid	.3350
Oil of Turpentine	.4260

GASES.	At Constant Pressure.	At Constant Volume.
Air (at freezing point)	.2375	.1685
Oxygen	.2175	.1551
Nitrogen	.2438	.1727
Hydrogen	3.4090	2.4123
Superheated steam*	.4805	.346
Carbon Monoxide (CO)	.2479	.1758
Carbon Dioxide (CO_2)	.2170	.1535
Olefiant Gas	.4040	.173
Blast Furnace Gas	.2277	.....
Chimney Gases (approx.)	.240	.....

*The specific heat of superheated steam is variable. See page 93.

If heat be applied to a pound of ice there will be a rise in temperature until the freezing point, 32° F., is reached. The ice will then begin to melt, but the temperature of the mixture of ice and water will remain 32° F., as long as any particle of ice remains in it. Yet the melting process will absorb heat. The amount thus absorbed in changing the state of a pound of ice from ice at 32° F., to water at 32° F. is 144 B. T. U. This is the *latent heat of fusion* of ice. If the application of heat be continued the temperature of the water will rise, but it will now require about twice as many heat units to effect a rise of one degree as it did to accomplish the same rise in the ice. The reason is that the specific heat of water is 1.00, while that of ice is only .504. When the water has reached a point of 212° F., there is a further absorption of heat with no increase of temperature. Boiling occurs, and the heat absorbed is expended in transforming the water into steam. Water at atmospheric pressure cannot be heated beyond 212° F., and the steam which is formed is also at a temperature of 212° F., when the entire pound of water has been evaporated into steam, 965.8 B. T. U. have been used in the operation. This is the *latent heat of evaporation* of water.

Ebullition—The temperature of ebullition of any liquid, or its boiling point, may be defined as that stage in the addition

TABLE 4

BOILING POINTS AT ATMOSPHERIC PRESSURE (*Kent*)

(14.7 lbs. absolute per square inch.)

Ammonia	140° F.	Water	212° F.
Bromin	145	Average sea water	213.2
Alcohol	173	Saturated brine	226
Benzine	212	Mercury	676

of heat to the liquid at which the temperature of the liquid is no longer increased and the heat added is absorbed by converting the liquid into vapor. This temperature depends upon the pressure under which the liquid is evaporated; the greater the pressure the higher the temperature.

Heat of the Liquid—In the evaporation of any liquid, that quantity of heat which is absorbed in raising the temperature from 32° F. to the temperature of ebullition corresponding to the particular pressure at which the evaporation occurs, is the *sensible heat*, or the *heat of the liquid*.

Total Heat of Evaporation—The quantity of heat required to raise a unit weight of any liquid from the freezing point to a given temperature, and entirely to evaporate it at that temperature is the total heat of evaporation of the liquid for that particular temperature. It is the sum of the heat of the liquid, and the latent heat of evaporation.

To recapitulate: The heat added to a body is divided up as follows:

Total Heat = Heat to change the temperature + heat to separate the molecules + heat to overcome the external pressure resisting an increase of volume of the body.

In case of water which is converted into steam the total heat is divided as follows:

Total Heat = Heat to change the temperature of the water + heat to separate the molecules of the water + heat to overcome resistance to increase in volume of the steam.
= Heat of the liquid + inner latent heat + outer latent heat.
= Heat of the liquid + total latent heat of steam.
= Total heat of evaporation.

The Steam Tables, page 74 give the heat of the liquid and total latent heat through a wide range of temperatures.

Gases—When heat is added to gases there is no inner work to be done, hence the total heat is that required to change the temperature plus that required to do the outer work. If the gas is not allowed to expand, as in case of gases heated at constant volume, the entire heat added is that required to change the temperature only.

Mechanical Equivalent of Heat—The relation between heat and mechanical work has been experimentally determined by Joule, who found that the heat necessary to raise the temperature of one pound of water one degree Fahr. at its maximum density can perform work equal to the raising of 772 pounds one foot high. This relation between heat and mechanical work is called the mechanical equivalent of heat, or *Joule's equivalent*. The latest experimental determination of Rowland shows that the exact value is somewhat higher than 772, and 778 foot-pounds is now usually accepted as the correct mechanical equivalent.

Transfer of Heat—Heat may be communicated from one body to another in three different ways: viz., by radiation, conduction and convection. *Radiation* is the transfer of heat between bodies separated by a transparent medium. *Conduction* is the transfer or flow of heat from a hotter to a colder particle in contact with it. *Convection* is the transfer of heat caused by the rise of heated particles in a mass of liquid or gas. The transfer of heat from a furnace to the boiler takes place by radiation, convection and conduction and the heat is distributed through the mass of water by convection, but the exact laws governing these methods of transfer are unknown.

Temperature of Fire—The following table, compiled by M. Pouillet, will enable the approximate temperature of a fire to be judged by its appearance. The temperature is practically the same for all kinds of combustibles under similar conditions.

TABLE 5

APPEARANCE OF FIRE.	TEMPERATURE FAHR.
Red, just visible	977°
" dull	1290
" cherry, dull	1470
" " full	1650
" " clear	1830
Orange, deep	2010
" clear	2190
White heat	2370
" bright	2550
" dazzling	2730

Linear Expansion of Substances by Heat—To find the increase in the length of a bar of any material due to an increase of temperature, multiply the number of degrees of increase of temperature by the co-efficient for 100° (Table 6) and by the length of the bar, and divide by 100.

The expansion of metals per degree rise of temperature increases slightly as higher temperatures are reached, but for all practical purposes it may be assumed to be constant.

(1) Mercurial Pyrometer for temperatures up to 800° Fahrenheit.

(2) Expansion Pyrometer for temperatures up to 1,500° Fahrenheit.

(3) Melting points of metals which flow at various temperatures up to the melting point of platinum, 3,227° Fahrenheit.

(4) Le Chatelier's thermo-electric pyrometer for temperatures up to 2,900° F.

(5) Calorimetry for temperatures up to 2,000° Fahrenheit.

TABLE 6

LINEAR EXPANSION OF SOLIDS AT ORDINARY TEMPERATURES

(The tabular values are the fractional increase in length for a temperature increase of 100° Fahrenheit or Centigrade.)

NAME OF SUBSTANCE.	COEFFICIENT PER 100° FAHRENHEIT.	COEFFICIENT PER 100° CENTIGRADE.
Brass, (cast)	.00104	.00188
Brass, (wire)	.00107	.00193
Brick, (fire)	.0003	.0005
Copper,	.0009	.0017
Glass, (English Flint)	.00045	.00081
Glass, (French white lead)	.00048	.00087
Gold	.0008	.0015
Granite, (average)	.00047	.00085
Iron, (cast)	.0006	.0011
Iron, (soft forged)	.0007	.0012
Iron, (wire)	.0008	.0014
Lead	.0016	.0029
Mercury	.0033	.0060
Platinum	.0005	.0009
Sandstone	.0006	.0011
Silver	.0011	.002
Slate, (Wales)	.0006	.001
Water, (varies considerably with the temperature)	.0086	.0155

High Temperature Measurements—The temperatures to be dealt with in steam boiler practise range from those of ordinary air and steam to the temperatures of burning fuel. The gases of combustion, originally at the temperature of the furnace, cool down as they pass through each successive bank of tubes in the boiler, to nearly the temperature of the steam, resulting in a wide range of temperatures through which definite measurements are sometimes required.

Of the different methods devised for ascertaining these temperatures, five of the most important will be mentioned, viz.:

Mercurial Pyrometers—Mercury boils at 676° F. and atmospheric pressure, and for temperatures above 500° F. the ordinary mercurial thermometer cannot be used. For higher temperatures, up to 800° F., the space above the mercury is filled with nitrogen gas, and as the mercury expands, the gas is compressed, increasing the pressure and raising the boiling point. So constructed, mercurial pyrometers can be used for indicating temperatures not exceeding 800° F.

Flue-gas temperatures are nearly always taken with such thermometers. The bulb of the instrument should project into the

path of maximum velocity of the gases in order that the average temperature may be obtained, and before a reading is taken, it is necessary to keep the thermometer inserted in the flue socket from seven to fifteen minutes depending on conditions. Sometimes these thermometers are made so that they can be permanently attached to the wall of the breeching or flue.

This is the most accurate and by far the most preferable method of recording stack and uptake temperatures.

Expansion Pyrometers—Brass expands about 50% more than iron for the same increase in temperature; and for both the expansion is nearly proportional to the rise. In expansion pyrometers this phenomenon is utilized by enclosing a brass rod in an iron pipe, one end of the rod being rigidly attached to a cap at the end of the pipe, and the other end connected by a multiplying gear to a pointer moving around a graduated dial. The whole length of the pipe must be at a uniform temperature before the full amount of expansion is obtained. This, together with the fact that lost motion is likely to exist in the mechanism connected to the pointer, makes the expansion pyrometer unreliable; it is only when the instrument is thoroughly understood and carefully calibrated that it can be depended upon. Its action is anomalous; for instance, if it is allowed to cool after being exposed to a high temperature, the needle will rise higher before it begins to fall. Similarly, a rise in temperature is first shown by the instrument as a fall. The explanation is that the iron, being on the outside, heats or cools more quickly than the brass. The readings are, therefore, valueless unless both the brass and iron are known to be of the same temperature.

Melting Points of Metals—When an approximate temperature is sufficient it can be found by introducing into the furnace or flue various metals of known melting points. The more common metals form a series in which the respective melting points differ by less than 100° to 200° F. and by using these in order, the temperatures can be fixed between the melting points of some two of them. This method lacks accuracy, but it suffices very well for approximate determination of the temperatures of the furnace, and in different parts of the tubes of a boiler.

TABLE 7

APPROXIMATE MELTING POINTS OF METALS

Wrought Iron melts	at	about	2825°	Fah.
Steel (low carbon)	"	"	2600°	"
Steel (high carbon)	"	"	2400°	"
Cast Iron (white)	"	"	2200°	"
Cast Iron (grey)	"	"	2000°	"
Copper	"	"	1975°	"
Gun Metal	"	"	1700°	"
Zinc	"	"	764°	"
Antimony	"	"	940°	"
Lead	"	"	618°	"
Bismuth	"	"	514°	"
Tin	"	"	447°	"
Platinum	"	"	3230°	"
Gold	"	"	2056°	"
Silver	"	"	1788°	"
Aluminum	"	"	1172°	"

Thermo-Electric Pyrometers—When wires of two different metals are joined at one end and heated, an electromotive force will be set up between the free ends of the wires. Its amount depends upon the composition of the wires and upon the temperature. If a delicate galvanometer of high resistance be connected to the "thermal couple", as it is called, the deflection of the needle, after a careful calibration, will indicate the temperature very accurately.

In the thermo-electric pyrometer of Le Chatelier, the wires are platinum and a 10% alloy of platinum and rhodium, enclosed in porcelain tubes to protect them from the oxidizing influence of the furnace gases. The couple with its protecting tubes is called an "element". The elements are made in different lengths to suit conditions. It is not necessary for accuracy to expose the whole length of the element to the temperature to be measured, as the electromotive force depends only upon the temperature of the juncture at the closed end of the protecting tube. The galvanometer can be located at any convenient point, since the length of the wires leading to it occasions practically no error.

The advantages of the thermo-electric pyrometer are: accuracy over a wide range of temperature, continuity of readings, and the ease with which observations can be taken. Its disadvantages are high first cost and extreme delicacy.

Calorimetry—This method derives its name from that fact that the process is the same as the determination of the specific heat of a substance by the water calorimeter, with the exception that in one the temperature is known and the specific heat is required, while in the other the specific heat is known and the temperature is required. The temperature is found as follows: A given weight of some substance, such as iron, nickle, platinum, or fire-brick, is heated to the unknown temperature and then plunged into water and the rise in temperature noted.

If:

X = unknown temperature of substance,
w = weight of heated substance, lbs.
W = weight of water, lbs.
T = final temperature of water,
t = temperature rise, or difference between initial and final temperatures of water,
s = specific heat of cooled body,

Then: $$X = T + \frac{Wt}{ws} \qquad [1]$$

Table 8 gives the specific heats of some substances used in this method. For furnace temperature determination, the constants in the second column should be used. Specific heats increase with temperature, and authorities differ as to the amount.

TABLE 8

MEAN SPECIFIC HEATS

SUBSTANCE	ORDINARY TEMPERATURE	MEAN FOR HIGH TEMPERATURE
Platinum	.032	.038
Iron (cast)	.130	.180
Nickel	.109	.136
Fire-brick	.200	.260

Example—A piece of wrought iron bar, weighing one-half pound, is thrown into the furnace and heated to the temperature of the fire, and is then withdrawn and placed in a pail containing ten pounds of water. The original temperature of water was 60° F., and after the immersion of the iron, the temperature rose 20°. The temperature of the furnace by the formula was then $X = 60 + \frac{10 \times 20}{.5 \times .18} = 2282°$ F., the specific heat of iron, 0.18, being taken from the table.

This method is affected by many sources of error, or else requires so many refinements of measurement that its results are usually very approximate.

UNION STEEL CO., PITTSBURG, PA., OPERATING 25,000 H. P. OF STIRLING BOILERS

Air

Pure Air is a mixture of oxygen and nitrogen in following proportions: by volume 20.91 parts oxygen to 79.09 parts nitrogen; by weight 23.15 parts oxygen to 76.85 parts nitrogen. Air in nature always contains other constituents such as dust, carbon dioxide, ammonia, ozone and water vapor.

Air being perfectly elastic, the density of the atmosphere decreases in geometrical ratio with the altitude. This fact has an important bearing on proportions of furnaces and stacks located in high altitudes, as will later appear. The atmospheric pressure for different altitudes is given in Table 12*.

Weight and Volume of Air—These depend upon the pressure and temperature, as expressed in the formula

$$Pv=53.3\,T \qquad [2]$$

In which: P = *absolute* pressure in pounds per square *foot*, v = volume in cubic feet of one pound of air, and T = *absolute* temperature Fah. of the air.

The weight of one cubic foot of air will be evidently $\frac{1}{v}$ pounds.

Example: Required the volume in cubic feet of a pound of air under 60.3 lbs. per square inch gauge pressure, at 115° Fah. Here $p = 144 \times (14.7+60.3) = 10{,}800$. $T = 115+461 = 576$, hence $v=\frac{53.3\times576}{10800}=2.84$ cu. ft. The weight per cubic foot will be $\frac{1}{v}=\frac{1}{2.84}$ $=0.35$ lbs.

Table 9 gives weight and volume of air at different temperatures.

The above formula will hold good if in place of the constant 53.3 the following constants be substituted for each gas:

Oxygen = 48.257
Nitrogen = 54.926
Hydrogen = 770.322

Specific Heat of Air—This varies with the temperature. At 266° it is two per cent., and at 446° it is 5.68 per cent., higher than at 32°, but the percentage of increase for such temperatures as exist in the boiler furnace, and along the path of the gases after they leave the furnace, is not known.

Vapor in Air—Air may carry as much as 3% of vapor. This fact is of considerable importance in solving problems relating to heating, drying and air compressing. Accordingly Table 10 gives the amount of vapor required to saturate air at different temperatures, its weight, expansive force, etc.

TABLE 9

VOLUME AND WEIGHT OF AIR AT VARIOUS TEMPERATURES, AND ATMOSPHERIC PRESSURE

TEMPERATURE IN DEGREES FAHR.	VOLUME OF ONE POUND CU. FT.	WEIGHT OF ONE CUBIC FT. IN LBS.
50	12.840	.077884
55	12.964	.077133
60	13.090	.076400
65	13.216	.075667
70	13.342	.074950
75	13.467	.074260
80	13.593	.073565
85	13.718	.072894
90	13.845	.072230
95	13.970	.07158c
100	14.096	.070942
110	14.346	.069698
120	14.598	.068500
130	14.849	.067342
140	15.100	.066221
150	15.352	.065140
160	15.603	.064088
170	15.854	.063072
180	16.106	.062090
190	16.357	.061134
200	16.606	.060210
210	16.860	.059313
212	16.910	.059135
220	17.111	.058442
230	17.362	.057596
240	17.612	.056774
250	17.865	.055975
260	18.116	.055200
270	18.367	.054444
280	18.621	.053710
290	18.870	.052994
300	19.121	.052297
320	19.624	.050959
340	20.126	.049686
360	20.630	.048476
380	21.131	.047323
400	21.634	.046223
425	22.262	.044920
450	22.890	.043686
475	23.518	.042520
500	24.146	.041414
525	24.775	.040364
550	25.403	.039365
575	26.031	.038415
600	26.659	.037510
650	27.913	.035822
700	29.172	.034280
750	30.428	.032865

*See page 58.

TABLE 10

WEIGHTS OF AIR, VAPOR OF WATER, AND SATURATED MIXTURES OF AIR AND VAPOR AT DIFFERENT TEMPERATURES, UNDER THE ORDINARY ATMOSPHERIC PRESSURE OF 29.921 INCHES OF MERCURY.

Temperature Degrees Fahrenheit.	Volume of Dry Air at different Temperatures, the Volume at 32° being 1.000.	Weight of a Cubic Foot of Dry Air at the different Temperatures, (Pounds).	Elastic Force of Vapor in Inches of Mercury, (Regnault).	Mixtures of Air Saturated with Vapor.						Cubic Feet of Vapor from 1 lb. of Water at its own Pressure in Column 4.
				Elastic Force of the Air in the Mixture of Air and Vapor in Inches of Mercury.	Weight of Cubic Foot of the Mixture of Air and Vapor.			Weight of Vapor mixed with 1 lb. of Air, in Pounds.	Weight of Dry Air mixed with 1 lb. of Vapor, in Pounds.	
					Weight of the Air in Pounds.	Weight of the Vapor in Pounds.	Total Weight of Mixture in Pounds.			
1	2	3	4	5	6	7	8	9	10	11
0°	.935	.0864	.044	29.877	.0863	.000079	.086379	.00092	1,092.4	.....
12°	.960	.0842	.074	29.849	.0840	.000130	.084130	.00155	646.1	.....
22°	.980	.0824	.118	29.803	.0821	.000202	.082302	.00245	406.4	.....
32°	1.000	.0807	.181	29.740	.0802	.000304	.080504	.00379	263.81	3,289
42°	1.020	.0791	.267	29.654	.0784	.000440	.078840	.00561	178.18	2,252
52°	1.041	.0776	.388	29.533	.0766	.000627	.077227	.00810	122.17	1,595
62°	1.061	.0761	.556	29.365	.0747	.000881	.075581	.01179	84.79	1,135
72°	1.082	.0747	.785	29.136	.0727	.001221	.073921	.01680	59.54	819
82°	1.102	.0733	1.092	28.829	.0706	.001667	.072267	.02361	42.35	600
92°	1.122	.0720	1.501	28.420	.0684	.002250	.070717	.03289	30.40	444
102°	1.143	.0707	2.036	27.885	.0659	.002997	.068897	.04547	21.98	334
112°	1.163	.0694	2.731	27.190	.0631	.003946	.067046	.06253	15.99	253
122°	1.184	.0682	3.621	26.300	.0599	.005142	.065042	.08584	11.65	194
132°	1.204	.0671	4.752	25.169	.0564	.006639	.063039	.11771	8.49	151
142°	1.224	.0660	6.165	23.756	.0524	.008473	.060873	.16170	6.18	118
152°	1.245	.0649	7.930	21.991	.0477	.010716	.058416	.22465	4.45	93.3
162°	1.265	.0638	10.099	19.822	.0423	.013415	.055715	.31713	3.15	74.5
172°	1.285	.0628	12.758	17.163	.0360	.016682	.052682	.46338	2.16	59.2
182°	1.306	.0618	15.960	13.961	.0288	.020536	.049336	.71300	1.402	48.6
192°	1.326	.0609	19.828	10.093	.0205	.025142	.045642	1.22643	.815	39.8
202°	1.347	.0600	24.450	5.471	.0109	.030545	.041445	2.80230	.357	32.7
212°	1.367	.0591	29.921	0.000	.0000	.036820	.036820	Infinite	.000	27.1

Column 5 = barometer pressure of 29,921, minus the proportion of this due to vapor pressure from column 4.

Water

Pure Water is a chemical compound of one volume of oxygen (O) and two of hydrogen (H), and its chemical symbol is H_2O.

Weight—The weight of water depends upon its temperature. Its weight at four temperatures much used in physical calculations is as follows:

TEMPERATURE FAHRENHEIT	WEIGHT PER CUBIC FOOT	WEIGHT PER CUBIC INCH
At 32° or freezing point at sea level	62.418 lbs.	.03612 lbs.
At 39.1° or maximum density.	62.425 "	.036125 "
At 62° or standard temperature.	62.355 "	.03608 "
At 212° or boiling point at sea level	59.760 "	.03458 "

Compressibility—Water is but slightly compressible, hence for all practical purposes it may be considered non-compressible. The coefficient of compressibility ranges from 0.000040 to 0.000051 per atmosphere, at ordinary temperatures, decreasing however with an increase of temperature.

Pressure due to Head—The weight of water at standard temperature being 62.355 lbs. per cubic foot,* the pressure exerted by a column of any stated height may be determined; and conversely the height of the column producing any stated pressure can be computed.

Pressure in pounds per square foot = 62.355 × height of column in feet.

Pressure in pounds per square inch = .433 × height of column in feet.

Height of column in feet = Pressure in pounds per square foot ÷ 62.355.

Height of column in feet = Pressure in pounds per square inch ÷ .433.

TABLE 11

VOLUME AND WEIGHT OF DISTILLED WATER AT VARIOUS TEMPERATURES (*Buel*)

Temperature Fahrenheit	Relative Volume, Water at 39.1° = 1	Weight per Cubic Foot. Pounds.	Temperature Fahrenheit	Relative Volume, Water at 39.1° = 1	Weight per Cubic Foot. Pounds.	Temperature Fahrenheit	Relative Volume, Water at 39.1° = 1	Weight per Cubic Foot. Pounds.	Temperature Fahrenheit	Relative Volume, Water at 39.1° = 1	Weight per Cubic Foot. Pounds.
32°	1.000129	62.42	160°	1.02324	61.01	290°	1.08405	57.59	430°	1.18982	52.47
39.1	1.000000	62.43	170	1.02671	60.80	300	1.09023	57.26	440	1.19898	52.07
40	1.000004	62.42	180	1.03033	60.59	310	1.09661	56.93	450	1.20833	51.66
50	1.000253	62.41	190	1.03411	60.37	320	1.10323	56.58	460	1.21790	51.26
60	1.000929	62.37	200	1.03807	60.14	330	1.11005	56.24	470	1.22767	50.85
70	1.001981	62.30	210	1.04226	59.89	340	1.11706	55.88	480	1.23766	50.44
80	1.00332	62.22	212	1.04312	59.71	350	1.12431	55.52	490	1.24785	50.03
90	1.00492	62.12	220	1.04668	59.64	360	1.13175	55.16	500	1.25828	49.61
100	1.00686	62.00	230	1.05142	59 37	370	1.13042	54.79	510	1.26892	49.20
110	1.00902	61.87	240	1.05633	59.10	380	1.14729	54.41	520	1.27975	48.79
120	1.01143	61.72	250	1.06144	58.81	390	1.15538	54.03	530	1.29080	48.36
130	1.01411	61.56	260	1.06679	58.52	400	1.16366	53.64	540	1.30204	47.94
140	1.01690	61.39	270	1.07233	58.21	410	1.17218	53.26	550	1.31354	47.52
150	1.01995	61.20	280	1.07809	57.90	420	1.18090	52.86	. . .	. . . .	. . . . .

Impurities and Solvent Power—Water in its natural state is never found absolutely pure. The composition of feed water to be used for boilers is of vital importance, the

*Authorities differ concerning the weight of water. At 62° F. the range is from 62.291 to 62.360, and 62.355 is generally accepted as the most accurate.

impurities existing in such water affecting not only the economy, but also the durability of the boiler. In solvent power water has a greater range than any other liquid. For common salt this is nearly constant at all temperatures, while it increases with an increase of temperature for such impurities as magnesium and sodium sulphate.

Sea water contains on an average about 3.125 per cent. part of its weight of solid matter, whose composition will be approximately:

Sodium Chloride	76%
Magnesium Chloride	10
Magnesium Sulphate	6
Calcium Sulphate or Gypsum	5
Calcium Carbonate	0½
Other Substances	2½
Total	100%

The following are the Boiling Points and Specific Gravities of sea water of varying density:

PERCENTAGE OF SALT	BOILING POINT FAHRENHEIT	SPECIFIC GRAVITY
3.125*	213.2°	1.029
6.250	214.4°	1.058
9.375	215.5°	1.087
12.500	216.7°	1.116
15.625	217.9°	1.145
18.750	219.1°	1.174

The boiling point of water decreases as the altitude above sea level is increased, as shown in Table 12.

Specific Heat of Water—The specific heat of water is unity at 39.1° F., but for other temperatures it is slightly greater. Rankine has constructed from Regnault's data the formula: Specific heat =

$$1+0.000000309\,(t-39.1)^2 \qquad [3]$$

In which t is the temperature Fah. In consequence of this variation of specific heat the *heat of the liquid* above 32° F. in any case is not exactly $t-32$, but is equal to $t-32+0.000000103\times(t-39.1)^3$, where t is the temperature of ebullition. The heat of the liquid as computed for several temperatures by both methods is given below:

TEMP. FAH.	T − 32	RANKINE'S FORMULA
60	28 B. T. U.	28.12 B. T. U.
100	68 " " "	68.01 " " "
150	118 " " "	118.30 " " "
200	168 " " "	168.70 " " "
212	180 " " "	180.79 " " "

It will thus be seen that the variation is entirely too slight to affect any but the most refined physical investigations, and that for all steam engineering work the heat of the liquid may be safely taken as $t-32$.

TABLE 12

BOILING POINT OF WATER AT VARIOUS ALTITUDES

Boiling Point in degrees Fahrenheit.	Altitude above Sea-Level. Feet.	Atmospheric Pressure. Pounds per square inch.	Barometer. Inches.
184	15,221	8.19	16.79
185	14,649	8.37	17.16
186	14,075	8.56	17.54
187	13,498	8.75	17.93
188	12,934	8.94	18.32
189	12,367	9.13	18.72
190	11,799	9.33	19.13
191	11,243	9.53	19.54
192	10,685	9.74	19.96
193	10,127	9.95	20.39
194	9,579	10.16	20.82
195	9,031	10.38	21.26
196	8,481	10.60	21.71
197	7,932	10.82	22.17
198	7,381	11.05	22.64
199	6,843	11.28	23.11
200	6,304	11.52	23.59
201	5,764	11.76	24.08
202	5,225	12.01	24.58
203	4,697	12.25	25.08
204	4,169	12.51	25.59
205	3,642	12.77	26.11
206	3,115	13.03	26.64
207	2,589	13.29	27.18
208	2,063	13.57	27.73
209	1,539	13.84	28.29
210	1,025	14.12	28.85
211	512	14.41	29.42
212	Sea-Level	14.70	30.00

*Or one thirty-second part of the weight of the water and the salt held in the solution.

Impurities in Boiler Feed Water

Natural waters usually contain other substances in solution or suspension. When the water is converted into steam, these substances, if solids, must be deposited somewhere in the boiler; if gases, they will pass out with the steam. The amount of solids deposited in a boiler is often astonishing; over 300 pounds per month will deposit in a 100 H. P. boiler using water containing only 7 grains per gallon. In the southwestern part of the United States where the water is often particularly bad, cases are known where boilers can be operated only two to three days between cleanings.

The treatment of feed water belongs to the chemist rather than to the engineer, hence when the water causes trouble it will be economy to submit the case to a *competent* chemist who makes a specialty of treating feed waters. His advice should be followed, since there are few cases where guessing is less successful than when treating feed waters. The following article is intended to convey such general information as will enable the reader to understand the effect of the impurities usually encountered, and to realize the necessity of referring the more difficult cases to an expert.

Effect of Impurities—According to the nature of the impurity it may produce one or several of the following results:

(1) Precipitation of mud, etc.

(2) Formation of scale.

(3) Formation of scum which causes excessive priming or foaming.

(4) Internal corrosion of the boiler.

Effect of Mud and Scale—Where provision is made to catch the mud and blow it off before it settles on the heating surface, the only evil effect is the loss of heat due to blowing off. If the mud is carried along and deposited on the heating surfaces it may unite with the scale-forming matter, and the mass will be baked to a hardness which renders its removal difficult and costly. The arrangement of feed and mud drum in the Stirling boiler is particularly efficacious in precipitating mud and silt, and the boiler is successfully operating on waters which many other types of boiler are unable to use.

The effect of scale depends largely upon its density. The scale formed by the carbonates is usually soft and porous, and its retarding effect upon heat transmission is small, hence unless present in large quantities its influence toward lowering boiler efficiency is less than commonly supposed. Sulphates and some other impurities form scales which are so hard that they can be removed only by cutting them loose, and so dense that they are impervious to water. Such scales are a source of positive danger which increases with the degree of temperature of the surface upon which they have formed, because the metal overheats and is liable to burn, crack, or bulge, thereby causing a rupture. The economy of the boiler is also seriously affected.

Scale-forming Materials—Those which occur most often and in largest quantity are

Calcium carbonate (lime)	$CaCO_3$
Magnesium carbonate	$MgCO_3$
Calcium sulphate	$CaSO_4$
Magnesium sulphate	$MgSO_4$

The following are less frequently found, and usually in small amounts:

Iron carbonate	Fe_2CO_3
Magnesium chloride	$MgCl_2$
Calcium chloride	$CaCl_2$
Potassium chloride	KCl
Sodium chloride	NaCl

and, variously, iron oxide and hydroxide, calcium phosphate, silica, and organic matter.

The carbonates of calcium and magnesium are but slightly soluble in water; they are usually combined with carbon dioxide as bicarbonates, $CaH_2(CO_3)_2$ and $MgH_2(CO_3)_2$ respectively, which are quite soluble in cold water. Heating the water drives off the carbon doxide, CO_2, and the bicarbonates decompose, precipitating, in the case of calcium the monocarbonate, $CaCO_3$, and magnesium hydrate, $Mg(OH)_2$, in the case of magnesium. This occurs between the temperatures 180° to 290° F. As the scale formed by calcium carbonate is porous and does not adhere strongly to the metal, it is not particularly troublesome unless present

in large quantity. The same is true of magnesium carbonate, but in the process of freeing the carbon dioxide, CO_2, the bicarbonate generally reduces to magnesium hydrate, $Mg(OH)_2$, which not only follows the water currents and settles very slowly, but it cements together such other matter as it may encounter. The violent foaming which is often caused by carbonates of calcium and magnesium may cause far more trouble than the scale produced by these salts.

The sulphates of calcium and magnesium are the most troublesome scale-forming impurities. They remain in solution until a temperature of about 302° is reached, when the calcium sulphate deposits in long, needle-like crystals, possessing active cementing properties. These mingle with any other matter present to form a hard resisting scale. The magnesium sulphate will deposit as a monohydrated salt, $MgSO_4H_2O$, and its presence is objectionable because it interferes with the removal of other impurities.

The iron carbonate behaves much like calcium monocarbonate, $CaCO_3$, but it occurs so seldom and in such small quantities that its presence has little effect.

The magnesium chloride precipitates as a hydroxide, which is objectionable because of its cementing properties. The other chlorides, calcium, potassium and sodium (common salt), give little trouble from incrustation unless allowed to concentrate beyond the point of saturation, when they are deposited and increase the bulk of the scale, although possessing no cementing properties themselves. They may, however, cause foaming, which will be greater as the solution becomes more concentrated. Frequent blowing off prevents their concentration, but wastes heat carried off by the hot water.

A temperature of 302° F. corresponds to 55 lbs. gauge pressure, hence the reason why the rear bank of the Stirling boiler removes most of the scale-forming matter before the hotter parts of the boiler are reached is now evident. The water upon entering the feed drum is heated to the temperature corresponding to the boiler pressure, hence during its course through the rear tubes the scale will be deposited and the purified water will pass into the hottest tubes in the front bank. This feature is peculiar to the Stirling boiler. Here also, as in case of feed water heaters, the element of time affects the degree to which the impurities can be removed, hence it is evident that the more the boiler is forced, the shorter the time the water can remain in the rear bank, hence the smaller the degree to which this water can be purified before passing into the front and middle banks.

TABLE 13

SOLUBILITIES OF SCALE-FORMING MINERALS

	SOLUBLE IN PARTS OF PURE WATER AT 32° F.	SOLUBLE IN PARTS OF CARBONATED WATER, COLD.	SOLUBLE IN PARTS OF PURE WATER AT 212°	INSOLUBLE IN WATER AT
Calcium Carbonate	62,500	150	62,500	302° F.
Calcium Sulphate	500	...	460	302° F.
Magnesium Carbonate	5,500	150	9,600	...
Calcium Phosphate	.....	1,333	.....	212° F.
Oxide of Iron	......	.....	......	212° F.
Silica	......	.....	......	212° F.

Scum may be due to vegetable matter, sewage, and light flocculent matter which gathers on top of the water. These form a glutinous skin on the water, which is raised by the steam, causing foaming or priming which may be so excessive as seriously to interfere with operation of the engines. When animal and vegetable oils are used as components of the cylinder oil, and the hot-well water containing them

is mixed with feed water containing soda, the oils are saponified, soap suds are formed, and violent foaming may result. The remedy is to use mineral oil. The scums are best handled by a surface blow-off.

Pitting and Corrosion are usually caused by free acids which are either originally in the water or are liberated by splitting up some salt. The acids may be of vegetable origin derived from some adulterant of mineral oil, or the original water may have been polluted with acid due to discharge from industrial works, or drainage from mines; waters from swamps and bogs often contain humic or vegetable acids; sulphuric acid is often absorbed from the atmosphere, and found in drainage from coal and ore mines, particularly if the ores are sulphides.

Magnesium chloride is generally thought to have a corrosive effect on boiler plate. Some assert that it is broken up by the boiling water into magnesium hydrate and hydrochloric acid; while others notably H. Ost, maintain that water is partly decomposed by boiling, and the iron of the boiler is attacked by liberated oxygen, the magnesium chloride subsequently combining with the iron protoxide thus formed; the reaction being as follows:

$$MgCl_2 + Fe(OH)_2 = FeCl_2 + Mg(OH)_2$$

If Ost's theory is correct, corrosion takes place quite independently of the magnesium chloride.

Air absorbed by water is liberated by boiling, and produces corrosion. The peculiar activity of air under such circumstances is due to the fact that whereas ordinary air is a mixture of oxygen and nitrogen in the approximate ratio of 1 to 4, air dissolved in water is a mixture of 1 part of oxygen and only 1.87 parts of nitrogen, owing to the greater solubility of oxygen. The deterrent effect of nitrogen as a dilutant is thus reduced, and the mixture is correspondingly more active chemically. In the experiments of Lt. Comdr. W. F. Worthington, U. S. N., samples of iron and steel were supported on glass rolls in a porcelain bath of distilled water through which air was forced. The corrosion of the iron and steel was marked, in some cases occurring uniformly over the surface of the samples, and in other cases being confined to pits of small area, but of surprising depth. The temperature at which the oxygen most rapidly attacks the iron is lower than that of steam at the pressures now used, hence it will be found that pitting will occur much faster in a boiler that is moderately warm than when in full service; it also occurs in places where the circulation is defective, such as in water-legs. The rapid corrosion of feed pipes is similarly explained—the temperature falls within the range in which oxygen rapidly attacks the iron.

When water contains alkali, any copper used in the boiler will rapidly pit and corrode in parts where the circulation is defective. The oxygen attacks the copper and the alkali dissolves the copper oxide so formed, thus presenting a fresh surface to the action of the oxygen. With such waters copper fire-box plates one-half inch thick have been pitted through in five months.

How to Handle Impure Feed Water—There are three courses of procedure, viz.:

(1) To neutralize the acids and remove the solids before the water enters the boiler.

(2) To treat the water with chemicals after it has entered the boiler, with a view of minimizing or preventing formation of incrustation and scale.

(3) To evaporate the water and remove at regular intervals the deposits which form.

Unless the water is of very good quality the first course is preferable, and the most economical in the end. The design of the equipment for efficient treatment will depend upon the character of the impurities in the water, and the work should be entrusted only to an expert. The necessary equipment is generally too expensive to be provided for small steam plants, and recourse must be had to the other methods.

Free acids should receive attention before the water enters the boiler or heater. While milk of lime fed into the boiler will form a thin coating which to some extent prevents the acid from corroding the metal, the proper procedure is to neutralize the acid before the water is used. If free acid only is present, it may be neutralized by addition of carbonate of soda, but an excess of the carbonate may cause considerable priming. If scale-forming materials are present with the acid, more elaborate processes will be necessary.

A VIEW FROM SHIPPING YARD WORKS OF THE STIRLING COMPANY, BARBERTON

If the water is suspected of containing acid, it should be tested by inserting blue litmus paper, which will turn red if acid is present. The water should then be submitted to a chemist, and the quantity of bicarbonate of soda required, or the necessity for more elaborate treatment, should be determined and prescribed by him.

Heating Feed Water not only saves heat, but serves as a means of external purification more or less efficient according to the kind of impurities present. At the temperature of 208° to 210° attainable in open or closed heaters some of the carbonates and other impurities can be precipitated. Since sulphate of lime precipitates at 302°, corresponding to 55 lbs. pressure, a live steam heater will remove most of it, if it be of *sufficient size and kept clean*. The element of time has considerable effect in precipitating impurities, hence the results will be better when both open and closed heaters are of sufficient size to allow the water to remain a considerable time under influence of the heat.

Treatment after Water Enters the Boiler—The object of such treatment is to convert the scale-forming impurities into others which are less objectionable. This method affords a fertile field to the vendor of "compounds." When prepared by a chemist for a particular water, such preparations may be of great benefit, but their use without adequate knowledge of what they contain and the effect of the ingredients on the impurities of the water, can be compared only with the use of "cure-alls" in medicine. When improperly used they produce quite as much trouble as the impurities they are expected to neutralize. Even when properly compounded their office is to convert a certain amount of objectionable solids into a *greater amount* of less objectionable solids. If they fail they have only increased the evil they were expected to cure, hence the necessity of consulting a chemist when compounds are to be used.

The Reagents used to Neutralize the Principal Impurities will now be given.

Calcium and magnesium carbonates are precipitated to some extent by heat, because the carbon dioxide, CO_2, necessary to hold them in solution is driven off. They may also be precipitated by caustic lime $CaOH_2$ which reduces them to the practically insoluble carbonates. Sodium hydroxide or caustic soda, NaOH, accomplishes the same result.

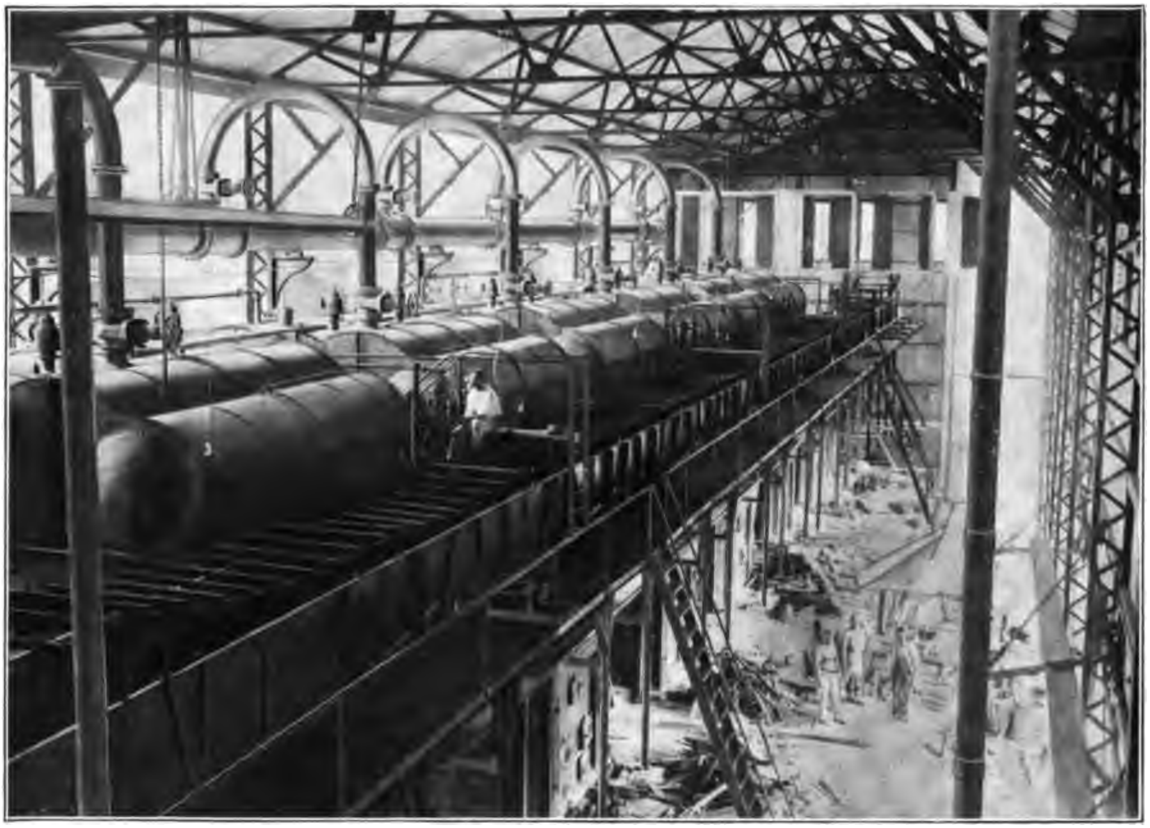
Magnesium and calcium sulphates and chlorides are convertible into carbonates by the use of soda-ash, Na_2CO_3. As carbonates, they are merely lesser evils than when sulphates. The resulting sodium sulphate or chloride, is harmless, requiring only occasional blowing off to prevent it from concentrating.

If the use of a solvent is necessary, there is in most plants no way of getting it into the boilers without shutting them down and introducing it through the manhole. If this is done only when the boilers are opened for cleaning no extra expense is involved, but, as a rule, the stoppages are so far apart that the introduction of the solvent accomplishes little, because in the natural course of running it will be blown out long before the next charge is introduced. Small quantities of solution introduced at short intervals are more effective than a large quantity at longer intervals, and when the water is very bad a much greater quantity of the solvent can be used in a given time than is possible where large quantities are introduced at long intervals. With some waters a charge of 30 pounds of soda-ash once a month might cause serious priming immediately after being introduced, while one pound a day could have no evil effect.

The proper way to introduce the solvent is to attach to the feed pump suction an apparatus arranged to feed it just as cylinder oil is fed to an engine. There are many inexpensive appliances of this kind on the market; a satisfactory home-made appliance consists of a tee in the suction pipe, with a gate valve on the hot-well side of it, the outlet of the tee being turned upward, and connected to a nipple which in turn is connected to a vessel containing the solution. A valve is also placed in the nipple. By opening this valve and closing the valve in suction pipe, the solution is drawn into the pump and passed to the boiler. The action must be intermittent, and is not so efficient as when a continuous feed in small quantity is used.

Removal of Scale—Even with a system of purification it is practically impossible to

get the water *absolutely* pure, hence some deposit will form in course of time Without purification this deposit will form more or less rapidly according to character of the water, and its removal is essential to efficiency and durability of the boiler. The most efficient appliance for this work is a turbine tube cleaner, the construction of which is described in the chapter on Boiler Cleaning.



4,000 H. P. OF STIRLING BOILERS, GUANICA CENTRALE, PUERTO RICO

Extraction of Oil and Grease from Feed Water—If feed water is taken from a hot-well the cylinder oil should be removed from it before it enters the boiler. In some districts where oil wells are common, much of the water available for boiler feed contains native oil. If this is allowed to enter the boilers, it will deposit on the hot surfaces, and these will inevitably be burned or blistered. The oil is also liable to cause heavy priming. "The peculiarity of grease deposits in boilers is that their effect is out of all proportion to their thicknesses. We have seen that scale of $\frac{1}{8}$-inch thickness will raise the temperature of the furnace plates about 300 degrees Fahrenheit. As grease offers ten times more resistance to heat, one would expect that .0125 inch would have the same effect as this thickness of scale, but experience shows that the merest trace of grease, certainly less than 0.001 inch, or one-tenth of the above, can cause far more injury than scale."*

Where the cylinder oil forms an emulsion with the hot-well water its removal is difficult, if not impossible, and the remedy is to select an oil which will not produce such emulsion. An oil extractor should be placed on the exhaust pipe, but while this, if properly looked after, will remove a considerable portion of the oil, the amount which passes through it is too great to be allowed to enter the boiler. Various devices are employed to extract the remainder of this oil. If an open heater is used it should be provided with an oil extracting device of liberal

*See *Water-Softening*, by C. E. Stromeyer and W. B. Baron. *Engineering*, London, Dec. 25, 1903.

capacity. In some cases the water from condenser or open heater is made to flow into a large box divided into compartments by vertical partitions, the water passing over one partition, then under the next, then over the third, etc., until it reaches the final compartment from which it is drawn by the feed pump. In each compartment is placed a basket made of wire netting, and loosely packed with hay, excelsior, etc., through which the water must pass. Each basket can be removed independently, and be charged with fresh filtering material. Each compartment is provided with surface and bottom blow-off. Another method is to insert into the feed pipe between the pump and the boiler two receptacles so piped that the water can be pumped through either of them while the other is being cleaned; these are arranged so that the water has to pass through layers of sponges, burlap, or other filtering material, which can easily be renewed. Whatever arrangement be adopted, it is necessary to renew the filtering material at regular intervals, and to ascertain by frequent inspection of the boiler that practically all the oil is removed from the feed water.

When oil has been allowed to enter the boiler it should at once be removed in the manner prescribed on page 228.

Good and Bad Feed Water—It is difficult to judge of the quality of a feed water by the number of grains of solids per gallon, for the reason that whereas 50 grains of some soluble salt, such as sodium chloride, might be handled with success, only 8 grains of calcium sulphate might render the water unsuitable for boilers. The following classification rates waters according to the number of grains of incrusting solids (calcium carbonate, magnesium carbonate, magnesium chloride, etc.) per gallon.

Less than 8 grains*..... Very good
From 8 to 12 " ...... Good
" 12 to 15 " ...... Fair
" 15 to 20 " ...... Poor
" 20 to 30 " ...... Bad§
Greater than 30 grains.... Very bad

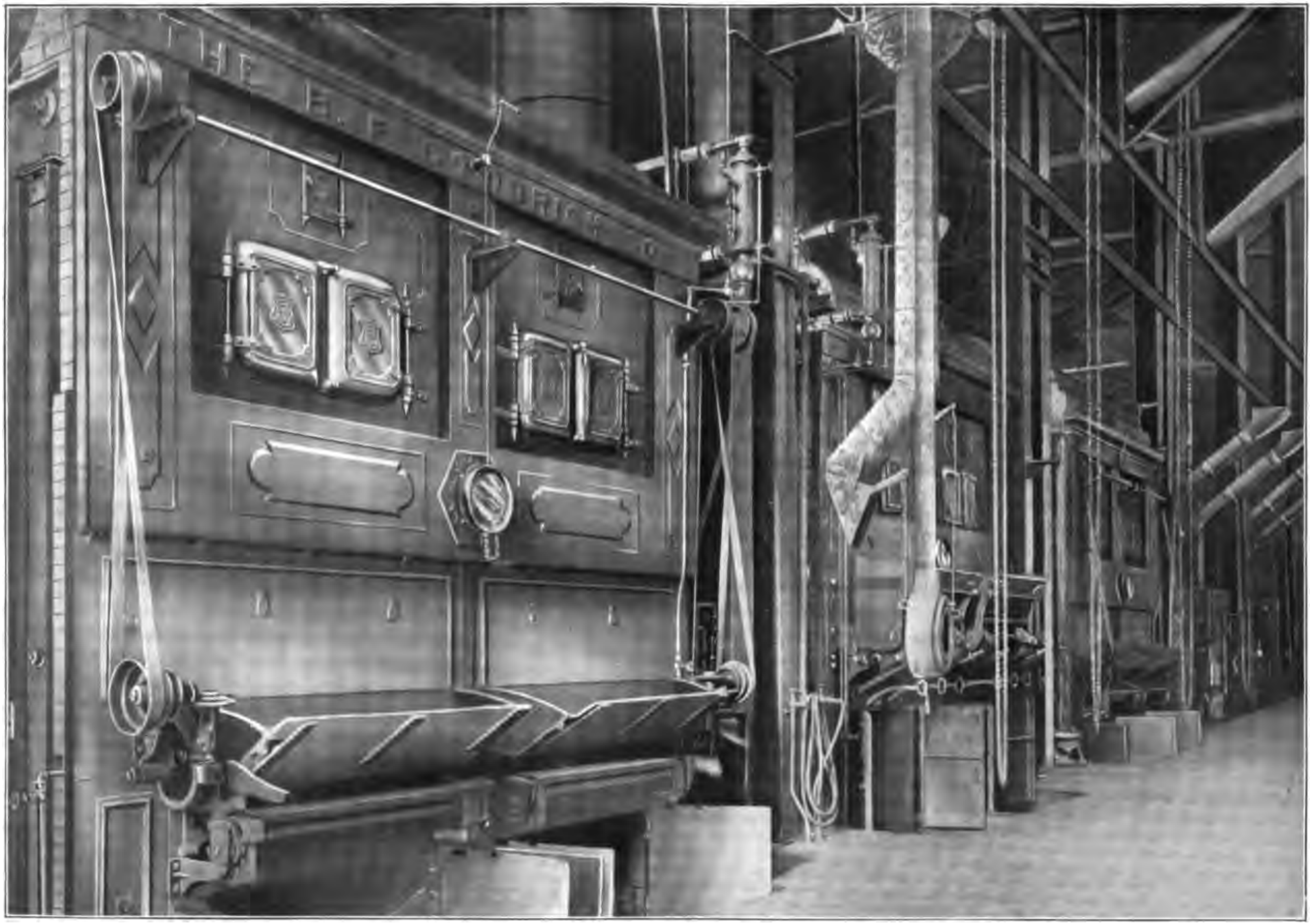
Much smaller quantities of sulphates of calcium (lime) and magnesium will bring the quality of water lower down in the scale. For a rough comparison it may be said that one grain of these solids would be as harmful as two to three grains of the carbonates and chlorides above mentioned.

TABLE 14

EFFECT OF, AND CORRECTIVES FOR, IMPURITIES IN FEED WATER (*Norton*)

TROUBLESOME SUBSTANCE.	TROUBLE.	REMEDY OR PALLIATION.
Sediment, mud, clay, etc. . . .	Incrustation.	Filtration. Blowing off.
Readily soluble salts	Incrustation.	Blowing off.
Bicarbonates of lime, magnesia, iron	Incrustation.	Heating feed. Addition of caustic soda, lime, or magnesia, etc.
Sulphate of lime	Incrustation.	Addition of carbonate of soda, barium chloride.
Chloride and sulphate of magnesium	Corrosion.	Addition of carbonate of soda.
Carbonate of soda in large amounts	Priming.‡	Addition of barium chloride.
Acid	Corrosion.	Alkali.
Dissolved carbonic acid and oxygen	Corrosion.	Heating feed. Addition of caustic soda or slacked lime.
Grease (from condensed water) .	Corrosion.	Slacked lime and filtering. Carbonate of soda. Substitute mineral oil.
Organic matter (sewage)	Priming.	Precipitate with alum or ferric chloride and filter.
Organic matter	Corrosion.	Precipitate with alum or ferric chloride and filter.

*One pound avoirdupois = 7,000 grains. ‡Similar result is caused by carbonates of calcium and magnesium. §Such water should not be used in a boiler unless first purified.



3,000 H. P. OF STIRLING BOILERS, B. F. GOODRICH RUBBER CO., AKRON OHIO

The Heating of Boiler Feed Water

Before the water fed into a boiler can be converted into steam it must be heated from its original temperature to that corresponding to the steam pressure. Steam at 160 lbs. gauge pressure has a temperature of about 370° Fahr., hence if the feed should be at a temperature of 60 degrees each pound of this water must absorb about 310 B. T. U. before it can be converted into steam. This amount of heat is nearly 27% of the total heat needed. Obviously, then, if before the water is pumped into the boiler it can be made to absorb heat which otherwise would go to waste through the engine exhaust, or the flue-gases, it will be economy to save this heat, provided the cost of doing so is less than the value of the heat which is saved.

The steam pressure and feed water temperatures before and after heating being known, the fuel saving can be computed by the following formula:

$$\text{Fuel saving in per cent.} = \frac{100\,(t-t_1)}{H+32-t_1} \qquad [4]$$

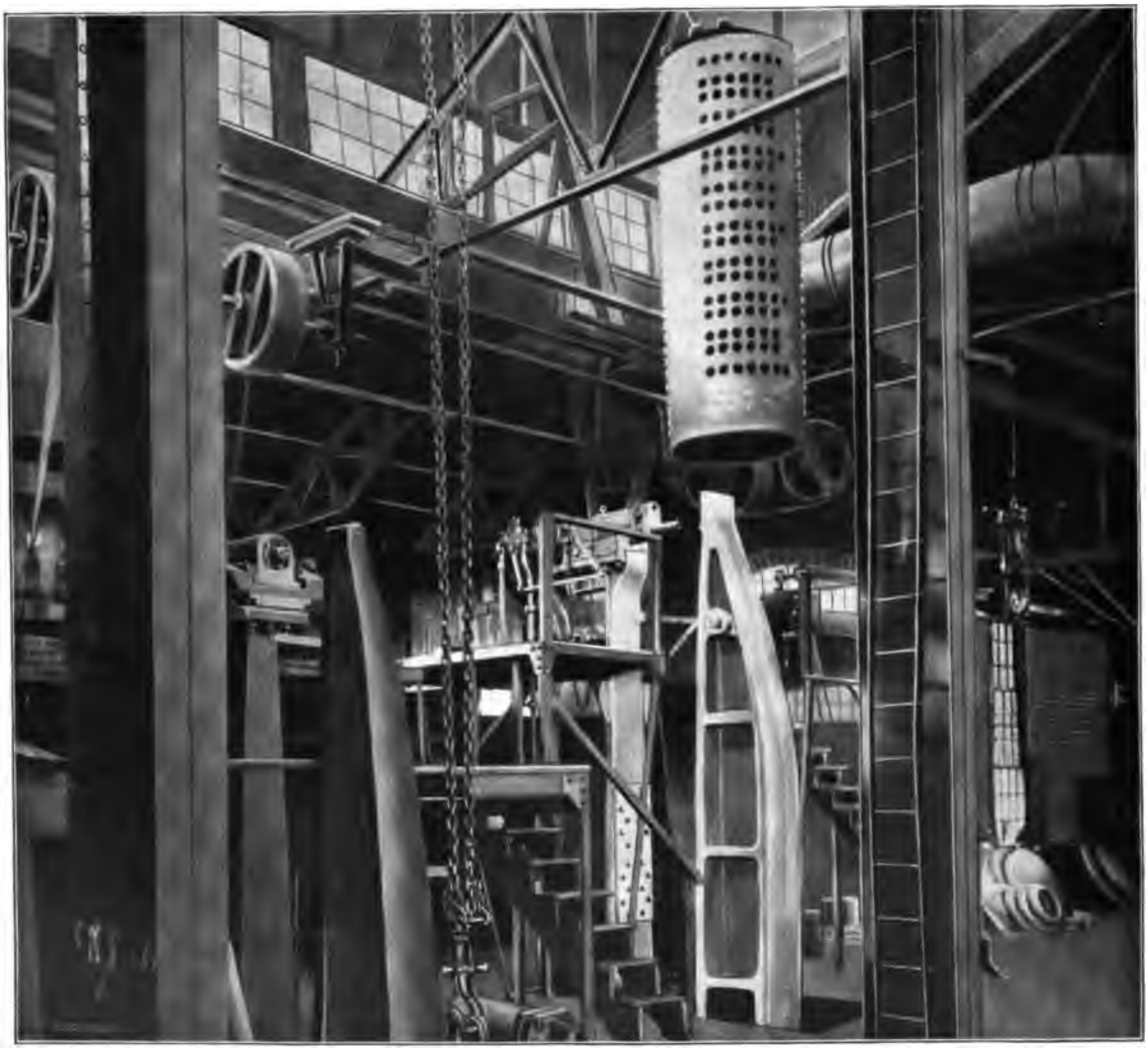
in which t = temperature Fahr. of feed water after heating, t_1 = temperature Fahr. of feed water before heating, and H = total B. T. U. above 32° Fahr. per. pound of steam at the boiler pressure from Table 18, page 74.

To effect this saving, money must be expended for feed heating apparatus, piping, space in which to install them, and labor to operate them. The heating may be done by use of exhaust steam heaters, of either the open or closed type, according to character of the feed water, and nature of the plant; or by economizers, or by a combination of the two systems. Which of these to choose can be determined only after a study of the conditions in each case. For example, if the exhaust steam can all be used for heating, drying, ice-making, etc., its value when so utilized may exceed its value as a feed heating medium, and an economizer should be considered. If the exhaust steam cannot be thus utilized, an open or closed heater can be considered, and here again the wisest selection can be made only after study of the conditions. When using certain feed waters heavily impregnated with mineral a closed heater may scale up so rapidly that its efficiency quickly falls off, and its cost of cleaning may be prohibitive, hence for such waters an open heater should be preferred. When engines work intermittently, as a mine hoist, a closed heater is not advisable, because the frequent coolings between hoists and the sudden heating when each lift begins will soon loosen the tubes, or even pull them apart, hence an open heater or an economizer will give more satisfactory service

Economizers are bulky, require a large amount of extra brickwork or an expensive metal housing, and frequently a considerable increase in the space necessary for heaters of the exhaust steam type. When computing the *net* return on an economizer investment, all these factors must be included. When the feed water is of a character that will quickly scale or incrust the economizer, and throw it out of service for cleaning during an excessive proportion of time, consideration must be given to the problem of purifying the water before it passes to the economizer, or the latter may fail to earn a profit on the investment. The character of fuel and type of boiler used also have more influence on the economizer problem than commonly supposed. The more wasteful the boiler, the greater the saving by using an economizer. When oil fuel is used under a large boiler of efficient design, the boiler efficiency may and often does exceed 80 per cent., thus leaving small opportunity for an economizer, and there are cases where economizers have been a source of profit while the boilers were fired with coal, but the *net saving* disappeared as soon as oil fuel was used, and the same would be the case with gas fuel.

From the foregoing it is evident that general data as to the saving that can be effected by heating feed water, or detailed computations based on assumed conditions, can be of little practical use. Each case must be independently worked out, which can be done intelligently only after exhaustive study of each of the conditions affecting that case, including probable life and growth of the plant. When as a result of such a study

the different methods practicable for handling the problem have been determined, the selection of the best one can be easily determined by balancing the saving possible with each method against its first cost, depreciation, maintenance, and cost of operation.



BULL RIVETERS IN DRUM SHOP OF THE STIRLING COMPANY'S WORKS

Thus far, only fuel saving has been considered. But beside this, the benefits of heating feed water are many. The proper office of a boiler is to *generate steam*, and to do this most profitably it requires to be kept clean. By properly heating the feed water, most of the impurities can be eliminated to the advantage of the boiler; the capacity of the boiler is increased by the amount of heat added to the feed, and the stresses caused by feeding cold water into the boiler are reduced. Introduction of cold water into a boiler is also an occasional cause of priming. For these reasons, heating of feed water is common even when fuel saving is not the main object, as in saw mills and in cases where feed water purification is necessary, as exemplified in the use of live steam heaters which purify, but considered as *heaters only*, save little or no fuel.

Steam

Gases and Vapors—If the temperature of a gas be kept constant its pressure may be increased by making a corresponding decrease in its volume. Vapors, which are simply liquids converted into aeriform condition, can exist only in connection with a definite pressure corresponding to each temperature. Thus water vapor, or *steam*, of 150 pounds per sq. inch absolute pressure, can exist only under a temperature of about 358° F.; hence if the pressure of saturated steam be fixed its temperature also becomes fixed, and vice versa.

Saturated Steam is steam which is at the maximum pressure and density possible at its temperature, or is steam in the condition in which it is generated from water with which it is in contact. If either the pressure be increased, or the temperature be decreased, some of the steam will immediately condense. Just so long as steam is of the same pressure and temperature as the water with which it can remain in contact without gaining or losing heat, it will remain saturated.

Quality of Steam—*Dry* saturated steam contains no water. In all practical cases saturated steam contains some water, which is suspended in it, and the steam is then said to be wet. The percentage weight of the steam, in a mixture of steam and water, is called the *quality of the steam*. Thus if it be found that for each 100 pounds of the mixture there is ¾-pound of water the quality of steam will be 99.25.

Superheated Steam—If heat be added to steam out of contact with water, both temperature and pressure increase, provided the volume remains constant, or the temperature and volume increase if the pressure remains constant. Steam whose temperature exceeds that of saturated steam of the same pressure is called superheated steam and its properties approximate those of a perfect gas.

Heat of the Liquid, Latent Heat, and Total Heat of Steam—As explained in the chapter on Heat, the heat necessary to raise the water from 32° F. to point of ebullition is called *heat of the liquid*. The heat absorbed during the ebullition consists of that necessary to dissociate the molecules, or the *inner latent* heat; and that necessary to overcome the resistance to increase in volume, or the *outer latent heat*, and these two are the total *latent heat of vaporization*, as given in the Steam Tables, page 74.

Relative Volume—The relative volume of steam is the ratio between the volumes of an equal weight of steam, and of water at 39.1° F., and it is equal to the volume in cubic feet of one pound of steam multiplied by 62.425. Example: A pound of steam at 250° F. occupies a volume of 13.65 cubic feet, hence its relative volume is

$$13.65 \times 62.425 = 852.1.$$

The Specific Volume of saturated steam is the volume in cubic feet of one pound of steam.

Boiling Point—The temperature of the boiling point of any liquid depends upon the pressure on the liquid. This fact is of great practical importance in steam condensers, and in many operations requiring boiling in an open vessel, since in the latter case altitude has considerable influence. The relation between altitude and boiling point is shown in Table 12, page 58.

Equivalent Evaporation from and at 212° F.—When comparing boiler tests, fuel performances, etc., it is usually found that neither the steam pressure nor the feed water temperature was the same in the various trials, hence it is necessary to establish some common basis to which all the trials can be reduced for purposes of comparison. The method of doing this is to transform the evaporation, as determined under the actual feed temperature and steam pressure noted during the test, into an equivalent evaporation based upon a standard feed water temperature of 212° F., and a steam pressure equal to normal atmospheric pressure at sea level (14.7 lbs. absolute). Under these standard conditions the steam will be generated *at* a temperature of 212° *from* water at 212°. The number of pounds of water which would be evaporated under the standard conditions by precisely the same amount of heat absorbed by the

TABLE 16

FACTORS OF EVAPORATION

STEAM PRESSURES, POUNDS BY GAUGE.*

Temp. F. of Feed Water	50	60	70	80	90	100	110	120	130	140	150	160	170
32	1.214	1.216	1.220	1.222	1.225	1.227	1.229	1.231	1.232	1.234	1.236	1.237	1.239
40	1.206	1.209	1.212	1.214	1.216	1.219	1.220	1.222	1.224	1.226	1.227	1.229	1.230
50	1.195	1.197	1.201	1.204	1.206	1.208	1.210	1.212	1.214	1.215	1.217	1.218	1.220
60	1.185	1.188	1.191	1.193	1.196	1.198	1.200	1.202	1.203	1.205	1.207	1.208	1.210
70	1.175	1.178	1.180	1.183	1.185	1.187	1.189	1.191	1.193	1.194	1.196	1.197	1.199
80	1.164	1.167	1.170	1.173	1.175	1.177	1.179	1.181	1.183	1.184	1.186	1.187	1.189
90	1.154	1.157	1.160	1.162	1.165	1.167	1.169	1.170	1.172	1.174	1.176	1.177	1.179
100	1.144	1.147	1.150	1.152	1.154	1.156	1.158	1.160	1.162	1.164	1.165	1.167	1.168
110	1.133	1.136	1.139	1.142	1.144	1.146	1.148	1.150	1.152	1.153	1.155	1.156	1.158
120	1.123	1.126	1.129	1.131	1.133	1.136	1.138	1.140	1.141	1.143	1.145	1.146	1.147
130	1.113	1.116	1.118	1.121	1.123	1.125	1.127	1.129	1.130	1.132	1.134	1.136	1.137
140	1.102	1.105	1.108	1.110	1.113	1.115	1.117	1.119	1.120	1.122	1.124	1.125	1.127
150	1.091	1.095	1.098	1.100	1.102	1.104	1.106	1.108	1.110	1.111	1.113	1.115	1.116
160	1.081	1.084	1.087	1.090	1.092	1.094	1.096	1.098	1.100	1.101	1.103	1.104	1.106
170	1.070	1.074	1.077	1.079	1.081	1.083	1.085	1.087	1.089	1.091	1.092	1.094	1.095
180	1.060	1.063	1.066	1.069	1.071	1.073	1.075	1.077	1.079	1.080	1.082	1.083	1.085
190	1.050	1.053	1.056	1.058	1.060	1.063	1.065	1.066	1.068	1.070	1.071	1.073	1.074
200	1.039	1.043	1.045	1.048	1.050	1.052	1.054	1.056	1.058	1.059	1.061	1.063	1.064
210	1.029	1.032	1.035	1.037	1.040	1.042	1.044	1.046	1.047	1.049	1.051	1.052	1.053
220	1.019	1.022	1.024	1.027	1.029	1.031	1.033	1.035	1.037	1.039	1.040	1.042	1.043
230	1.008	1.011	1.014	1.017	1.019	1.021	1.023	1.025	1.027	1.028	1.030	1.031	1.033
240	0.9077	1.001	1.004	1.006	1.008	1.011	1.013	1.014	1.016	1.018	1.019	1.021	1.022
250	0.9873	0.9904	0.9909	0.9956	0.9979	1.000	1.002	1.004	1.006	1.007	1.009	1.010	1.012
260	0.9768	0.9799	0.9826	0.9851	0.9874	0.9896	0.9915	0.9934	0.9952	0.9968	0.9985	0.9999	1.001
270	0.9663	0.9695	0.9722	0.9747	0.9769	0.9791	0.9811	0.9820	0.9848	0.9863	0.9881	0.9895	0.9910
280	0.9560	0.9501	0.9618	0.9643	0.9666	0.9688	0.9707	0.9725	0.9743	0.9759	0.9777	0.9791	0.9806
290	0.9455	0.9486	0.9514	0.9539	0.9561	0.9583	0.9603	0.9621	0.9639	0.9655	0.9673	0.9687	0.9702
300	.....	0.9382	0.9409	0.9434	0.9457	0.9485	0.9408	0.9517	0.9534	0.9551	0.9568	0.9582	0.9597

Temp. F. of Feed Water	180	190	200	210	220	230	240	250	260	270	280	290	300
32	1.240	1.241	1.243	1.244	1.245	1.246	1.247	1.248	1.250	1.251	1.252	1.253	1.254
40	1.232	1.233	1.234	1.236	1.237	1.238	1.239	1.240	1.241	1.242	1.243	1.244	1.245
50	1.221	1.223	1.224	1.225	1.226	1.228	1.229	1.230	1.231	1.232	1.233	1.234	1.235
60	1.211	1.212	1.214	1.215	1.216	1.217	1.218	1.219	1.220	1.221	1.222	1.223	1.224
70	1.200	1.202	1.203	1.205	1.206	1.207	1.208	1.209	1.210	1.211	1.212	1.213	1.214
80	1.190	1.192	1.193	1.194	1.195	1.196	1.198	1.199	1.200	1.201	1.202	1.203	1.204
90	1.180	1.181	1.183	1.184	1.185	1.186	1.187	1.188	1.189	1.190	1.191	1.192	1.193
100	1.170	1.171	1.172	1.174	1.175	1.176	1.177	1.178	1.179	1.180	1.181	1.182	1.183
110	1.159	1.160	1.162	1.163	1.164	1.166	1.167	1.168	1.169	1.170	1.171	1.172	1.173
120	1.149	1.150	1.151	1.153	1.154	1.155	1.156	1.157	1.158	1.159	1.160	1.161	1.162
130	1.138	1.140	1.141	1.142	1.144	1.145	1.146	1.147	1.148	1.149	1.150	1.151	1.152
140	1.128	1.129	1.131	1.132	1.133	1.134	1.135	1.136	1.137	1.138	1.139	1.140	1.141
150	1.118	1.119	1.120	1.121	1.123	1.124	1.125	1.126	1.127	1.128	1.129	1.130	1.131
160	1.107	1.108	1.110	1.111	1.112	1.113	1.115	1.116	1.117	1.118	1.119	1.120	1.121
170	1.097	1.098	1.099	1.101	1.102	1.103	1.104	1.105	1.106	1.107	1.108	1.109	1.110
180	1.086	1.088	1.089	1.090	1.091	1.093	1.094	1.095	1.096	1.097	1.098	1.099	1.100
190	1.076	1.077	1.078	1.080	1.081	1.082	1.083	1.084	1.085	1.086	1.087	1.088	1.089
200	1.065	1.067	1.068	1.069	1.071	1.072	1.073	1.074	1.075	1.076	1.077	1.078	1.079
210	1.055	1.056	1.057	1.059	1.060	1.061	1.062	1.063	1.064	1.065	1.066	1.067	1.068
220	1.045	1.046	1.047	1.048	1.050	1.051	1.052	1.053	1.054	1.055	1.056	1.057	1.058
230	1.034	1.035	1.037	1.038	1.039	1.040	1.041	1.043	1.044	1.045	1.046	1.047	1.047
240	1.024	1.025	1.026	1.028	1.029	1.030	1.031	1.032	1.033	1.034	1.035	1.036	1.037
250	1.013	1.015	1.016	1.017	1.018	1.020	1.021	1.022	1.023	1.024	1.025	1.026	1.027
260	1.003	1.004	1.005	1.007	1.008	1.009	1.010	1.011	1.012	1.013	1.014	1.015	1.016
270	0.9924	0.9937	0.9949	0.9962	0.9974	0.9986	0.9997	1.001	1.002	1.003	1.004	1.005	1.006
280	0.9820	0.9833	0.9845	0.9858	0.9870	0.9882	0.9893	0.9904	0.9914	0.9924	0.9934	0.9944	0.9953
290	0.9715	0.9729	0.9741	0.9754	0.9766	0.9777	0.9789	0.9799	0.9800	0.9820	0.9830	0.9840	0.9849
300	0.9610	0.9624	0.9636	0.9649	0.9661	0.9672	0.9684	0.9694	0.9704	0.9715	0.9725	0.9735	0.9744

The values for intermediate pressures and feed water temperatures may, with sufficient accuracy for all practical purposes, be obtained by interpolation. If *exact* values are necessary they may be computed by the Formula

$$\text{Factor of Evaporation} = \frac{H - t + 32}{965.8}$$

In which H = total heat of steam above 32° from Table No. 18, and t = the temperature Fahrenheit of the boiler feed water.

*When using this table in connection with the Steam Table, No. 18, recollect that in the latter table the pressures are absolute, or gauge pressures + 14.7 lbs.

boiler under the actual test conditions is called the *equivalent evaporation from and at* 212°; the quotient of the equivalent evaporation from and at 212°, divided by the actual evaporation under test conditions, is the *factor of evaporation*. For example, suppose a boiler to evaporate water from a feed temperature of 60° F. into steam at 60 lbs. gauge pressure. The total heat above 32° in a pound of steam at 60 lbs. gauge (74.7 lbs. absolute) is 1175.6 B. T. U. But since the water was originally at 60° instead of 32°, the heat added by the boiler was only $1175.6-(60-32)=1147.6$ B. T. U. Had this same heat been used to evaporate steam at atmospheric pressure from water already at a temperature of 212°, only 965.8 B. T. U. per pound of water, or the latent heat of evaporation, would have been needed, hence for each pound of water the boiler evaporated under the actual conditions, it could have evaporated $\frac{1147.6}{965.8}=1.188$ lbs. of water from and at 212°.

Similarly, in another case it might be found that for each pound evaporated under the actual conditions the same heat would evaporate 1.121 pounds of water from and at 212°. The values 1.188 and 1.121 are the *factors of evaporation*, or the factor by which the number of pounds of water in the actual test is to be multiplied to find the equivalent number of pounds that could be evaporated from and at 212° F. with the same amount of heat. This factor for any set of

TABLE 15

PROPERTIES OF SATURATED STEAM FOR VARYING AMOUNTS OF VACUUM*

Pressure Above Vacuum. Lbs. per sq. in.	Vacuum in Inches of Mercury	Temperature, Degrees Fahrenheit.	Heat of the Liquid above 32° Fahrenheit, B. T. U.	Latent Heat above 32° Fahrenheit, B. T. U.	Total Heat above 32° Fahrenheit, B. T. U.	Density or Weight per Cubic Foot, Pounds.
.245	29½	58.8	26.84	1072.6	1099.5	0.00077
.490	29	79.3	47.40	1058.3	1105.7	0.00152
.735	28½	92.0	60.04	1049.6	1109.6	0.00223
.980	28	101.4	69.47	1043.1	1112.5	1.00294
1.47	27	115.3	83.36	1033.4	1116.8	0.00431
1.96	26	125.6	93.73	1026.1	1119.8	0.00566
2.45	25	133.9	102.1	1020.3	1122.4	0.00699
2.94	24	140.9	109.1	1015.5	1124.6	0.00829
3.43	23	147.0	115.3	1011.1	1126.4	0.00961
3.92	22	152.3	120.6	1007.4	1128.0	0.01087
4.41	21	157.2	125.6	1003.9	1129.6	0.01218
4.90	20	161.5	129.9	1000.9	1130.9	0.01342
5.39	19	165.6	134.0	998.1	1132.1	0.01468
5.88	18	169.4	137.9	995.3	1133.2	0.01594
6.37	17	172.8	141.3	992.9	1134.2	0.01719
6.86	16	176.0	144.5	990.7	1135.2	0.01839
7.35	15	179.1	147.6	988.5	1136.2	0.01963
7.84	14	182.1	150.6	986.4	1137.1	0.02087
8.82	12	187.5	156.1	982.7	1138.7	0.02334
9.80	10	192.4	161.0	979.2	1140.2	0.02576
12.25	5	203.1	171.8	971.7	1143.5	0.03178
14.69	0	212.1	180.9	965.3	1146.2	0.03765

*Partly from S. A. Reeve, *The Thermodynamics of Heat Engines*, 1903.

conditions may be determined by the formula:

$$F = \frac{H-t+32}{965.8} \qquad [5]$$

in which H=total heat of steam above 32° from steam table; t=temperature Fah. of feed water.

Table 16 gives the factors of evaporation for a large range of conditions, and for all except the most refined work the omitted values may be determined by interpolation.

A Unit of Evaporation is the quantity of heat necessary to evaporate one pound of water at 212° into steam at the same temperature, and is equal to 965.8 B. T. U. Its symbol is U. E.

Absolute and Gauge Pressures—Steam gauges indicate the pressure above the atmosphere. The atmospheric pressure changes according to altitude and the variations of the barometer. Consequently calculations involving the properties of steam are based on *absolute* pressure, which is equal to the gauge pressure plus the atmospheric pressure in pounds per square inch. The latter is usually assumed to be equal to 14.7 pounds per sq. inch at sea level,* but for other levels it must be determined from the barometer reading at that place.

Vacuum gauges indicate the difference, expressed in inches of mercury, between atmospheric pressure and the pressure inside the vessel to which the gauge is attached. For all rough purposes two inches height of mercury may be considered equal to pressure of one pound per square inch, hence for any reading of the vacuum gauge the

TABLE 17

RATE OF VARIATION OF PROPERTIES OF SATURATED STEAM AT VARIOUS PRESSURES

Absolute Pressure. Lbs. per sq. in.	Temperature. Deg. Fahr.	Increase in Temperature per Pound of Pressure. Deg. Fahr.	Heat of the Liquid. B. T. U.	Increase in Heat of the Liquid, per Pound of Pressure. B. T. U	Latent Heat. B. T. U.	Decrease in Latent Heat per Pound of Pressure. B. T. U.	Total Heat. B. T. U.	Increase in Total Heat per Pound of Pressure. B. T. U.	Per Cent. of Total Heat as Heat of Liquid.
20	227.9		196.9		954.6		1151.5		17.3
		1.96		1.98		1.38		.595	
40	267.1		236.4		927.0		1163.4		20.3
		1.27		1.28		.885		.390	
60	292.5		261.9		909.3		1171.2		22.4
		.965		.975		.685		.290	
80	311.8		281.4		895.6		1177.0		23.9
		.830		.825		.580		.245	
100	327.6		297.9		884.		1181.9		25.2
		.614		.642		.456		.186	
150	358.3		330.0		861.2		1191.2		27.7
		.468		.492		.348		.144	
200	381.7		354.6		843.8		1198.4		29.8
		.357		.373		.264		.109	
300	417.4		391.9		817.4		1209.3		32.4
		.275		.279		.195		.084	
400	444.9		419.8		797.9		1217.7		34.5
		.225		.237		.169		.068	
500	467.4		443.5		781.0		1224.5		35.5
		.179		.190		.135		.054	
750	512.1		490.9		747.2		1238.0		39.6
		.139		.149		.107		.043	
1000	546.8		528.3		720.3		1248.7		42.4

*See Table 12, page 58.

absolute pressure will be $14.7 - \frac{1}{2} \times$ gauge reading in inches. Example: vacuum of 24 inches will equal $14.7 - 12 = 2.7$ lbs. absolute per sq. inch.

Table 15 gives the temperature, pressure and other properties of steam for varying amounts of vacuum, and *exact* pressures corresponding to each inch of reading of vacuum gauge.

Economy of High Pressure Steam—From the steam tables the following condensed table of the heat needed at different pressures may be constructed.

ABSOLUTE PRESSURE.	TEMPERATURE F.	HEAT OF LIQUID.	LATENT HEAT.	TOTAL HEAT.
14.7	212	180.8	965.8	1146.6
20.0	228	196.9	954.6	1151.5
100.0	327.6	297.9	884.0	1181.9
301.9	418	392.5	816.9	1209.4

From this the following conclusions can be drawn.

As the pressure and temperature increase, the latent heat decreases, but less rapidly than heat of the liquid increases, hence the total heat increases. The percentage increase of total heat is very small, being for the pressures of 20, 100 and 301.9 pounds absolute, only 0.43, 3.0 and 5.4 per cent. respectively, more than required for the pressure of 14.7 lbs. The temperatures, however, increase at the rates of 7.5, 54.5 and 97.1 per cent. The efficiency for a perfect steam engine is proportional to the expression $\frac{t - t_1}{t}$, in which t and t_1 are absolute temperatures of steam at admission and exhaust, respectively. In actual engines the efficiency only approximates to the ideal, yet it will follow the same general law. Since the exhaust temperature cannot be lowered beyond present practise it follows that the only available method of increasing the efficiency is to raise the temperature at admission, which means either higher steam pressure, or use of superheated steam. As above shown, the increase in pressure will require but a trifling increase in fuel, hence the higher the pressure the greater the economy.

Steam Tables—Up to the present time an algebraic expression for the relation between saturated steam pressures, temperatures, and volumes, has not been produced, except empirically. These relations have, however, been experimentally dedetermined by Regnault, and from his data steam tables have been computed. These obviate the necessity of using empirical formulas. Such formulas may be found in standard works on Thermodynamics, and a number of them are given in Peabody's work below referred to. The following named tables cover all practical cases:

Table 15 gives properties of saturated steam for varying amounts of vacuum.

Table 17 shows variation in properties of steam at different pressures.

Table 18 gives properties of saturated steam from 2 to 500 pounds absolute. These tables are based partly on Prof. Cecil H. Peabody's *Tables of the Properties of Saturated Steam*, which are generally accepted by engineers.

TABLE 18

PROPERTIES OF SATURATED STEAM

*Pressure above Vacuum. Lbs. per sq. in.	Temperature, Degrees Fahrenheit.	Heat of Liquid above 32° Fahr. B. T. U.	Latent Heat above 32° Fahr. B. T. U.	Total Heat above 32° Fahr. B. T. U.	Density, or Weight per Cubic Foot. Pounds.
2	126.3	94.4	1026.1	1120.5	0.00576
4	153.1	121.4	1007.2	1128.6	0.01107
6	170.1	138.6	995.2	1133.8	0.01622
8	182.9	151.5	986.2	1137.7	0.02125
10	193.3	161.9	979.0	1140.9	0.02621
12	202.0	170.7	972.9	1143.6	0.03111
14	209.6	178.3	967.5	1145.8	0.03600
14.7	212.0	180.8	965.8	1146.6	0.03760
16	216.3	185.1	962.8	1147.9	0.04067
18	222.4	191.3	958.5	1149.8	0.04547
20	228.0	196.9	954.6	1151.5	0.05023
22	233.1	202.0	951.0	1153.0	0.05495
24	237.8	206.8	947.6	1154.4	0.05966
26	242.2	211.2	944.6	1155.8	0.06432
28	246.4	215.4	941.7	1157.1	0.06899
30	250.3	219.4	938.9	1158.3	0.07360
32	254.0	223.1	936.3	1159.4	0.07821
34	257.5	226.7	933.7	1160.4	0.08280
36	260.9	230.0	931.5	1161.5	0.08736
38	264.1	233.3	929.2	1162.5	0.09191
40	267.1	236.4	927.0	1163.4	0.09644
42	270.1	239.3	925.0	1164.3	0.1009
44	272.9	242.2	923.0	1165.2	0.1054
46	275.7	245.0	921.0	1166.0	0.1099
48	278.3	247.6	919.2	1166.8	0.1144
50	280.9	250.2	917.4	1167.6	0.1188
52	283.3	252.7	915.7	1168.4	0.1233
54	285.7	255.1	914.0	1169.1	0.1277
56	288.1	257.5	912.3	1169.8	0.1321
58	290.3	259.7	910.8	1170.5	0.1366
60	292.5	261.9	909.3	1171.2	0.1409
62	294.7	264.1	907.7	1171.8	0.1453
64	296.7	266.2	906.2	1172.4	0.1497
66	298.8	268.3	904.7	1173.0	0.1541
68	300.8	270.3	903.3	1173.6	0.1584
70	302.7	272.2	902.1	1174.3	0.1628
72	304.6	274.1	900.8	1174.9	0.1671
74	306.5	276.0	899.4	1175.4	0.1714
76	308.3	277.8	898.2	1176.0	0.1757
78	310.1	279.6	896.9	1176.5	0.1801
80	311.8	281.4	895.6	1177.0	0.1843
82	313.5	283.2	894.4	1177.6	0.1886
84	315.2	285.0	893.1	1178.1	0.1930
86	316.8	286.7	891.9	1178.6	0.1973
88	318.5	288.4	890.7	1179.1	0.2016

*To reduce to gauge pressures at sea level, subtract 14.7 from pressures in this column. In altitudes above sea level, subtract pressures per square inch as in Table 12, page 58.

PROPERTIES OF SATURATED STEAM.—CONTINUED

*Pressure above Vacuum. Lbs. Per sq. in.	Temperature, Degrees Fahrenheit.	Heat of Liquid above 32° Fahr. B. T. U.	Latent Heat above 32° Fahr. B. T. U.	Total Heat above 32° Fahr. B. T. U.	Density, or Weight Per Cubic Foot. Pounds.
90	320.0	290.0	889.6	1179.6	0.2058
92	321.6	291.6	888.4	1180.0	0.2101
94	323.1	293.2	887.3	1180.5	0.2144
96	324.6	294.8	886.2	1181.0	0.2186
98	326.1	296.4	885.0	1181.4	0.2229
100	327.6	297.9	884.0	1181.9	0.2271
102	329.0	299.4	882.9	1182.3	0.2314
104	330.4	300.9	881.8	1182.7	0.2356
106	331.8	302.3	880.8	1183.1	0.2399
108	333.2	303.8	879.8	1183.6	0.2441
110	334.6	305.2	878.8	1184.0	0.2484
112	335.9	306.6	877.8	1184.4	0.2526
114	337.2	308.0	876.8	1184.8	0.2568
116	338.5	309.4	875.8	1185.2	0.2610
118	339.8	310.7	874.9	1185.6	0.2653
120	341.1	312.0	874.0	1186.0	0.2695
122	342.3	313.3	873.0	1186.3	0.2736
124	343.5	314.6	872.1	1186.7	0.2779
126	344.7	315.9	871.2	1187.1	0.2820
128	345.9	317.1	870.3	1187.4	0.2862
130	347.1	318.4	869.4	1187.8	0.2904
132	348.3	319.6	868.6	1188.2	0.2946
134	349.5	320.8	867.7	1188.5	0.2988
136	350.6	322.0	866.9	1188.9	0.3030
138	351.7	323.2	866.0	1189.2	0.3072
140	352.9	324.4	865.1	1189.5	0.3113
142	354.0	325.6	864.3	1189.9	0.3155
144	355.1	326.7	863.5	1190.2	0.3197
146	356.1	327.8	862.8	1190.6	0.3239
148	357.2	328.9	862.0	1190.9	0.3280
150	358.3	330.0	861.2	1191.2	0.3321
152	359.3	331.1	860.4	1191.5	0.3363
154	360.3	332.2	859.6	1191.8	0.3405
156	361.4	333.3	858.9	1192.2	0.3447
158	362.4	334.3	858.2	1192.5	0.3488
160	363.4	335.4	857.4	1192.8	0.3530
162	364.4	336.4	856.7	1193.1	0.3572
164	365.4	337.5	855.9	1193.4	0.3614
166	366.4	338.5	855.2	1193.7	0.3655
168	367.3	339.5	854.5	1194.0	0 3695
170	368.3	340.5	853.8	1194.3	0.3737
172	369.2	341.5	853.1	1194.6	0.3778
174	370.2	342.5	852.3	1194.8	0.3820
176	371.1	343.5	851.6	1195.1	0.3862
178	372.1	344.4	851.0	1195.4	0.3904
180	373.0	345.4	850.3	1195.7	0.3945

*To reduce to gauge pressures at sea level, subtract 14.7 from pressures in this column. In altitudes above sea level, subtract pressures per square inch as in Table 12, page 58.

PROPERTIES OF SATURATED STEAM.—CONTINUED

*Pressure above Vacuum. Lbs. per sq. in.	Temperature, Degrees Fahrenheit.	Heat of Liquid above 32° Fahr. B. T. U.	Latent Heat above 32° Fahr. B. T. U.	Total Heat above 32° Fahr. B. T. U.	Density, or Weight per Cubic Foot. Pounds.
182	373.9	346.4	849.6	1196.0	0.3987
184	374.8	347.3	848.9	1196.2	0.4029
186	375.7	348.2	848.3	1196.5	0.4070
188	376.6	349.2	847.6	1196.8	0.4111
190	377.4	350.1	847.0	1197.1	0.4153
192	378.3	351.0	846.3	1197.3	0.4194
194	379.2	351.9	845.7	1197.6	0.4236
196	380.0	352.8	845.0	1197.8	0.4278
198	380.9	353.7	844.4	1198.1	0.4318
200	381.7	354.6	843.8	1198.4	0.4359
202	382.6	355.4	843.2	1198.6	0.4399
204	383.4	356.3	842.6	1198.9	0.4441
206	384.2	357.2	841.9	1199.1	0.4482
208	385.1	358.0	841.4	1199.4	0.4524
210	385.9	358.9	840.7	1199.6	0.4565
212	386.7	359.7	840.2	1199.9	0.4607
214	387.5	360.6	839.5	1200.1	0.4648
216	388.3	361.4	839.0	1200.4	0.4690
218	389.1	362.2	838.4	1200.6	0.4731
220	389.8	363.0	837.8	1200.8	0.4772
222	390.6	363.9	837.2	1201.1	0.4813
224	391.4	364.7	836.6	1201.3	0.4855
226	392.2	365.5	836.1	1201.6	0.4896
228	392.9	366.3	835.5	1201.8	0.4939
230	393.7	367.1	834.9	1202.0	0.4979
232	394.5	367.9	834.3	1202.2	0.5021
234	395.2	368.6	833.9	1202.5	0.5062
236	395.9	369.4	833.3	1202.7	0.5103
238	396.7	370.2	832.7	1202.9	0.5144
240	397.4	371.0	832.2	1203.2	0.5186
242	398.1	371.7	831.7	1203.4	0.5226
244	398.9	372.5	831.1	1203.6	0.5268
246	399.6	373.2	830.6	1203.8	0.5311
248	400.3	374.0	830.0	1204.0	0.5353
250	401.0	374.7	829.5	1204.2	0.5393
252	401.7	375.4	829.1	1204.5	0.5433
254	402.4	376.2	828.5	1204.7	0.5475
256	403.1	376.9	828.0	1204.9	0.5517
258	403.8	377.6	827.5	1205.1	0.5559
260	404.5	378.4	826.9	1205.3	0.5601
262	405.2	379.1	826.4	1205.5	0.5642
264	405.8	379.8	825.9	1205.7	0.5684
266	406.5	380.5	825.4	1205.9	0.5726
268	407.2	381.2	824.9	1206.1	0.5767
270	407.9	381.9	824.4	1206.3	0.5809
272	408.5	382.6	823.9	1206.5	0.5850

*To reduce to gauge pressures at sea level, subtract 14.7 from pressures in this column. In altitudes above sea level, subtract pressures per square inch as in Table 12, page 58.

PROPERTIES OF SATURATED STEAM.—Continued

*Pressure above Vacuum. Lbs. per. sq. in.	Temperature, Degrees Fahrenheit.	Heat of Liquid above 32° Fahr. B. T. U.	Latent Heat above 32° Fahr. B. T. U.	Total Heat above 32° Fahr. B. T. U.	Density, or Weight per Cubic Foot. Pounds.
274	409.2	383.3	823.4	1206.7	0.5892
276	409.8	384.0	822.9	1206.9	0.5934
278	410.5	384.6	822.5	1207.1	0.5976
280	411.1	385.3	822.0	1207.3	0.602
282	411.8	386.0	821.5	1207.5	0.606
284	412.4	386.6	821.1	1207.7	0.610
286	413.0	387.3	820.6	1207.9	0.614
288	413.7	388.0	820.1	1208.1	0.618
290	414.3	388.6	819.7	1208.3	0.622
292	414.9	389.3	819.2	1208.5	0.627
294	415.6	390.0	818.7	1208.7	0.631
296	416.2	390.6	818.3	1208.9	0.635
298	416.8	391.3	817.8	1209.1	0.639
300	417.4	391.9	817.4	1209.3	0.644
302	418.0	392.5	816.9	1209.4	0.648
304	418.6	393.2	816.4	1209.6	0.652
306	419.2	393.8	816.0	1209.8	0.656
308	419.8	394.4	815.6	1210.0	0.660
310	420.4	395.0	815.2	1210.2	0.664
312	421.0	395.7	814.7	1210.4	0.668
314	421.6	396.3	814.2	1210.5	0.673
316	422.2	396.9	813.8	1210.7	0.677
318	422.8	397.5	813.4	1210.9	0.681
320	423.4	398.1	813.0	1211.1	0.685
322	424.0	398.7	812.5	1211.2	0.690
324	424.5	399.3	812.1	1211.4	0.694
326	425.1	399.9	811.7	1211.6	0.698
328	425.7	400.5	811.3	1211.8	0.702
330	426.2	401.1	810.8	1211.9	0.707
335	427.6	402.6	809.8	1212.4	0.717
350	431.9	406.9	806.8	1213.7	0.748
375	438.4	414.2	801.5	1215.7	0.800
400	445.2	421.4	796.3	1217.7	0.853
450	456.2	433.4	787.7	1221.1	0.959
500	466.6	444.3	779.9	1224.2	1.065

*To reduce to gauge pressures at sea level, subtract 14.7 from pressures in this column. In altitudes above sea level, subtract pressures per square inch as in Table 12, page 58.

For relation between Heat of the Liquid, Latent Heat, and Total Heat, see page 50.

1,250 H. P. OF STIRLING BOILERS, CONSOLIDATED MAIN REEF MINES AND ESTATES, LTD., JOHANNESBURG, SOUTH AFRICA

Moisture in Steam

Practically all saturated steam contains water, varying in amount from a fraction of one per cent. when the steam is generated in a properly designed boiler fed with good water, to five per cent. or even more when the feed water is bad, or the boilers are of defective design. Not only is the heat absorbed by raising this water from the boiler feed temperature to the steam temperature practically wasted, but the water causes further loss by increasing the initial condensation in the engine cylinder; it also interferes with proper cylinder lubrication, causes knocking in the engine, and water hammer in the steam pipe.

Quality of Steam—The percentage weight of steam, in a mixture of steam and water, is called the quality of the steam. Thus steam of quality 99.5 contains one-half of one per cent. by weight of moisture.

Calorimeters—The apparatus used to determine the content of moisture in steam is called a calorimeter, though the name is inapt, since the instrument is in no sense a measurer of heat. The first form used was the "barrel calorimeter," in this apparatus liability of error is so great that its use is practically abandoned. Modern calorimeters are usually of either the throttling or separator type.

Throttling Calorimeter—Fig. 14 shows a section through a typical form of the instrument. Steam is drawn from the vertical pipe by a nipple arranged as later described, passes around the first thermometer cup as shown, then through a hole about $\frac{1}{8}$-inch diameter in the disk as shown. It next passes around the lower thermometer cup, after which it is permitted to escape. Thermometers are inserted into the cups, which are then filled with cylinder oil, and when the whole apparatus is heated the temperature of the steam before and after passing through the hole in the disk is noted.

The instrument and pipes leading to it should be thoroughly covered to diminish the radiation loss.

When steam passes from a higher to a lower pressure, as in this case, no work has to be done in overcoming a resistance; hence, assuming there is no loss from radiation, the quantity of heat is exactly the same after passing the disk as it was ahead of it. Suppose that the higher steam pressure is 150 lbs. by gauge, and the lower pressure that of the atmosphere. The total heat in a pound of dry steam at the former pressure is 1193.5 B. T. U. and at the latter pressure is 1146.6 B. T. U., difference, 46.9 B. T. U. As this heat still exists in the steam of lower pressure,

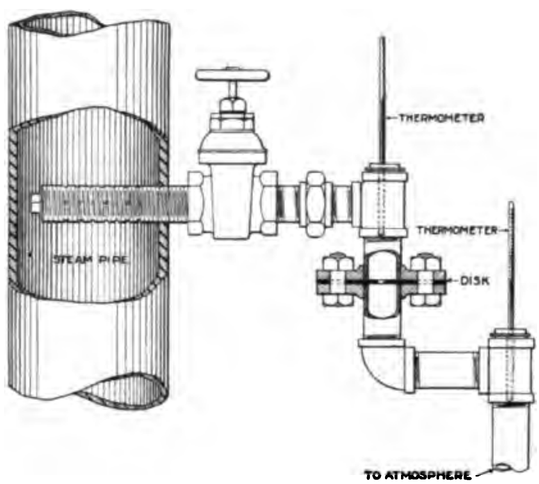


FIG. 14. THROTTLING CALORIMETER AND SAMPLING PIPE

its effect is to *superheat* that steam. Assuming the specific heat of steam to be 0.48, the steam will then be superheated $\frac{46.9}{0.48} = 97.7$ degrees. Suppose, however, the steam had contained one per cent. of moisture. Before any superheating could occur, this moisture would have to be evaporated into steam of atmospheric pressure. Since the latent heat of steam at atmospheric pressure is 965.8 B. T. U. it follows that the one per cent. of moisture would require 9.58 B. T. U. to evaporate it, leaving only $46.9 - 69.658 = 37.242$ B. T. U. available for superheating, hence the superheat would be $\frac{37.242}{0.48} = 77.6°$ as against 97.7 degrees in the preceding case. In a similar manner the degree of superheat for other amounts of moisture can be determined, and the action of the throttling calorimeter is based on this fact as will now be shown.

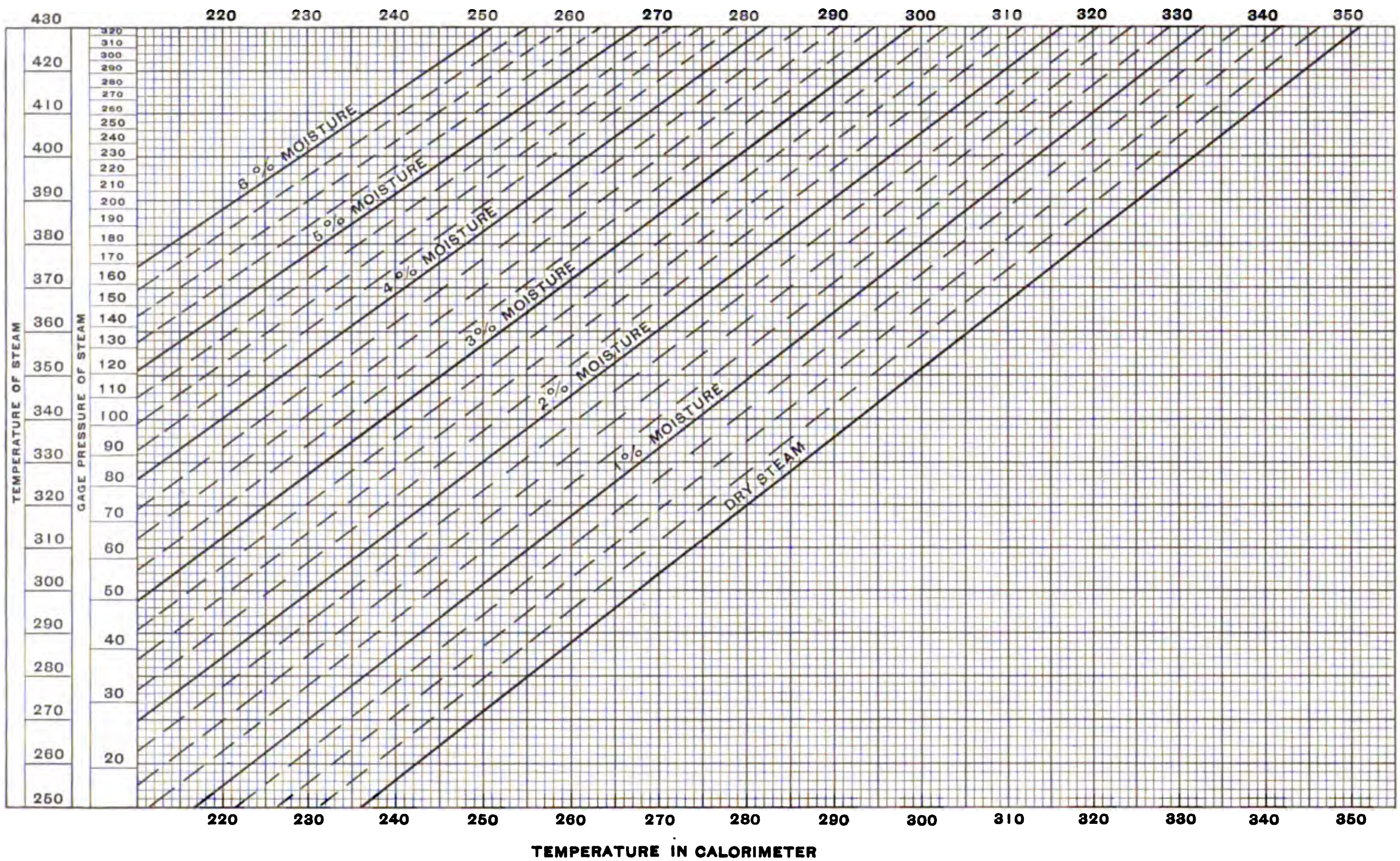


FIG. 15. CHART ENABLING RESULTS FROM THROTTLING CALORIMETER TO BE OBTAINED WITHOUT COMPUTATION BASED ON FORMULA NO. 7

Let H = total heat of steam at boiler pressure.

L = latent heat of steam at boiler pressure.

h = total heat of steam at reduced pressure after passing the disk.

t_1 = temperature of *saturated* steam at the reduced pressure.

t_2 = temperature of steam after expanding through opening in the disk.

0.48 = specific heat of saturated steam.

x = proportion of moisture in the steam.

The difference between the B. T. U.'s in a pound of steam at boiler pressure and after passing the disk is the heat which must evaporate the moisture in the steam, and then do the superheating, hence

$H-h=xL-0.48\ (t_2-t_1)$, therefore

$$x=\frac{H-h-0.48\ (t_2-t_1)}{L} \qquad [6]$$

Almost invariably the lower pressure is taken as that of the atmosphere where $h=1146.6$ and $t_1=212$, hence the formula becomes

$$x=\frac{H-1146.6-0.48\ (t_2-212)}{L} \qquad [7]$$

For practical work it is more convenient to dispense with the upper thermometer in the calorimeter, and substitute an accurate steam guage whose readings are more easily noted.

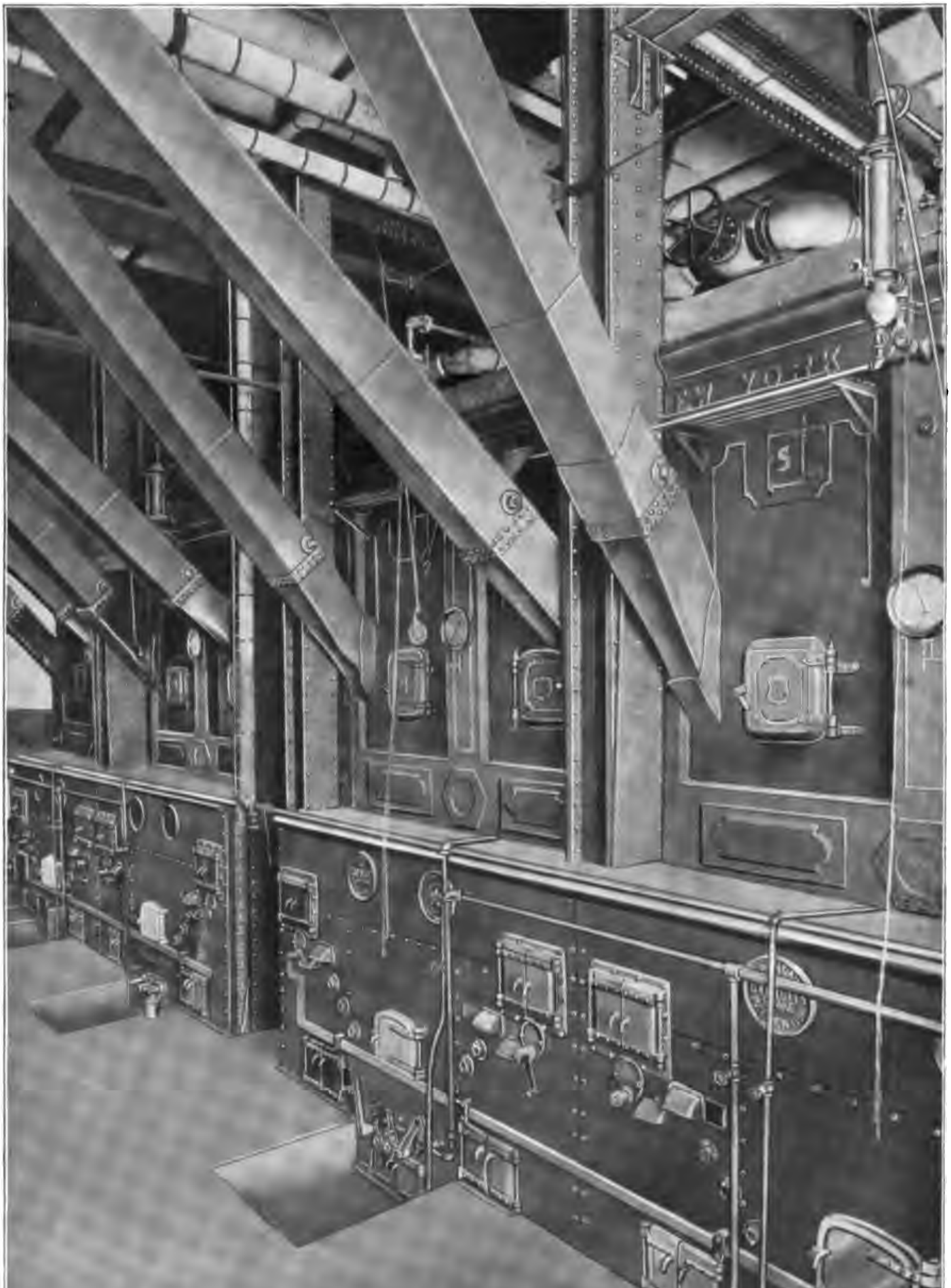
The value of x may be obtained, without computation, from Fig. 15. To illustrate its use, suppose that the steam gauge on the calorimeter indicated 160 pounds pressure, and that the temperature t_2 in the calorimeter after the steam had expanded was 304°. Look on the bottom line of the chart and locate the vertical line over 304°; look into the gauge pressure scale on the left side of the chart, and locate the horizontal line opposite 160 pounds; these two lines intersect the diagonal line indicating one-half of one per cent. of moisture. If instead of a steam gauge to indicate the pressure, a thermometer had been used to indicate the temperature of the steam before expanding, the temperature on the upper thermometer would have been 370°, and the column of temperatures on the extreme left hand of the chart would be the one to use; the horizontal line opposite 370° will be found to be the same one which is opposite 160 pounds pressure, hence as before the moisture will be one-half of one per cent.

Sources of Error—There are two. The first is that the specific heat of superheated steam, while given as 0.48, is far from being certain, and only future investigation can determine the true value. The second source of error is loss of heat by radiation. Evidently from the moment the steam enters the sampling nipple it is losing heat, hence when it passes through the small opening and into the lower pressure the heat available for evaporating moisture and superheating will be diminished by just the amount lost by radiation, hence the value of t_2 will be lower than it should be. This is sometimes corrected for as follows: A valve in the steam pipe beyond the calorimeter nipple is closed, and the steam left in a quiescent state for about ten minutes, and it is *assumed* that by doing this all the moisture in the steam will settle out, and that a sample of steam drawn from the pipe will be dry. Steam is then allowed to flow through the calorimeter and the temperature of the lower thermometer is noted. Let T denote this temperature. Since the sample of steam was assumed to be dry it follows that if there were no loss from radiation the value of T would be that due to all of the liberated heat being absorbed in superheating the steam of lower temperature. There is, however, a loss by radiation, and the effect of this is to condense some of the steam of lower pressure, and the water thus formed must be evaporated before any superheating can be done. Let x^1 represent the proportion of water thus formed, then evidently

$$x^1=\frac{H-h-0.48\ (T-t_1)}{L}$$

Now this amount of water was not in the steam originally, but was caused by condensation in the instrument, hence the *true* amount of moisture in the steam, which may be denoted by X, will be

$$X=x-x^1=\frac{H-h-0.48\ (t_2-t_1)}{L}-\frac{H-h-0.48\ (T-t_1)}{L}=\frac{0.48\ (T-t_2)}{L} \qquad [8]$$



JACOB RUPPERT ICE CO., NEW YORK, OPERATING 2,700 H. P. OF STIRLING BOILERS

The disadvantages of this method are: (1) It assumes that during the test the boiler pressure will remain the same as it was when T was determined, which is seldom practicable; (2) It assumes that the sample of steam drawn into the instrument when determining T was absolutely dry, although experiment has shown that this assumption is not necessarily true. Notwithstanding these facts, formula [8] is much used by engineers because of its simplicity and convenience, and any error due to its use is of no practical significance.

There are many forms of throttling calorimeter, all of which operate on precisely the same principle as the simple design shown in Fig. 14. An extremely convenient and compact design is shown in Fig. 16. It consists of two concentric cylinders screwed to a cap containing a thermometer cup. The steam pressure is measured by a gauge placed in the supply pipe, or any other convenient place.

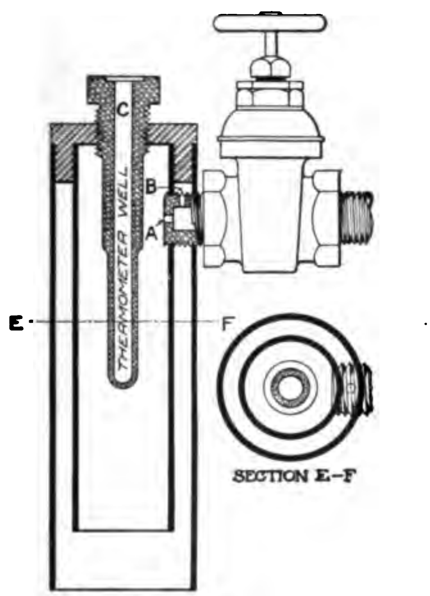


FIG. 16. COMPACT THROTTLING CALORIMETER

Steam passes through the opening A, expands to atmospheric pressure, and its temperature at this pressure is measured by a thermometer placed in the cup C. To prevent radiation losses the annular space between the two cylinders is used as a jacket, and is supplied with steam through the hole B.

The limits of the throttling calorimeter at sea level are from about four per cent. of moisture at eighty pounds pressure to six per cent. at 200 pounds pressure. If there is a greater content of moisture the liberated heat is insufficient to evaporate it and superheat the steam thus generated.

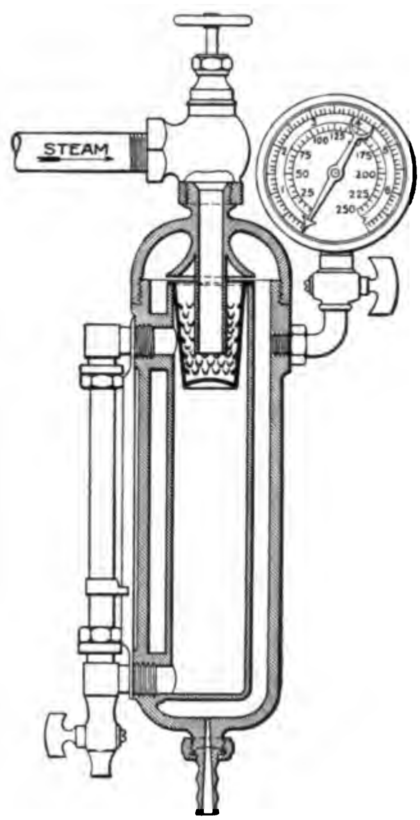
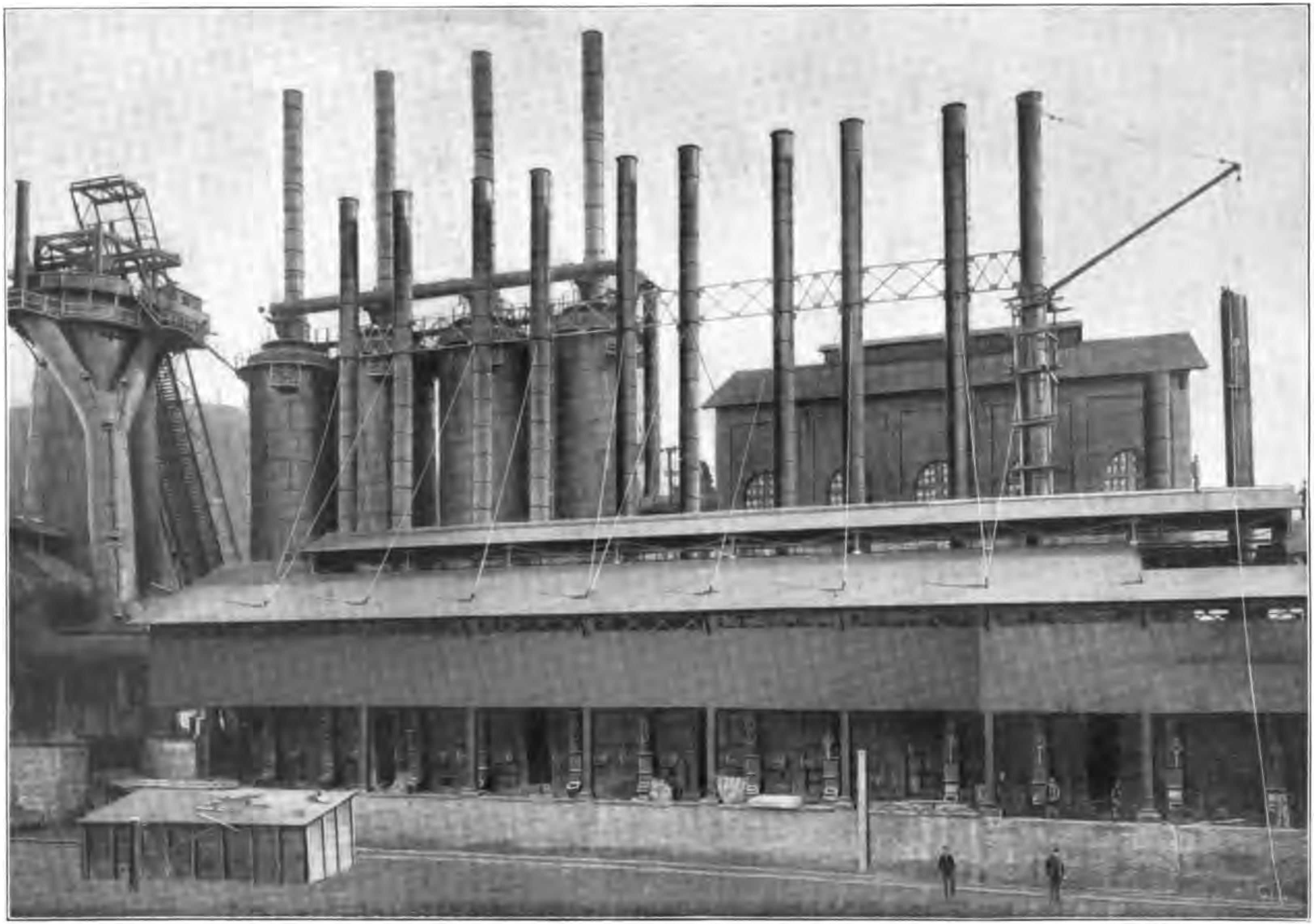


FIG. 17. SEPARATING CALORIMETER

Separating Calorimeter—The separating calorimeter mechanically separates the entrained water from the steam and collects it in a reservoir, where its amount is either indicated by a gauge glass or determined by draining it off and weighing it. The steam passes out of the calorimeter through an orifice of known size, so that either its total amount can be calculated or it can be weighed as later described. To avoid radiation errors the calorimeter should be well covered with non-conducting material. This instrument is not limited in capacity theoretically, but if the amount of moisture is very large, the readings should be checked by passing the discharged steam through a throttling calori-



3,700 H. P. OF STIRLING BOILERS, RIVERSIDE IRON WORKS, WHEELING, WEST VIRGINIA

meter; that is, a small separator should be used between the steam pipe and a throttling calorimeter, and the sum of the percentages obtained from the two instruments be taken as the moisture in the steam.

In the separating calorimeter, the amount of steam passing through the orifice can be determined by Napier's empirical formula, page 91. There is liability of considerable error in determining the area of such small orifices, and further, the flow of steam soon wears the orifice larger. A more accurate method of determining the weight of steam passing through is to convey it through a hose into a barrel of water resting on a platform scale. The weight of the barrel and contained water having been noted before and after the steam is run in, the difference is the weight of steam condensed. The moisture caught in the separating calorimeter can be weighed in the same way. If W is the weight of steam condensed, w the weight of moisture from the separating calorimeter, and x the per cent. of moisture in the steam, then

$$x = \frac{100w}{W + w} \qquad [9]$$

Location of Sampling Nipple—The principal source of inaccuracy in calorimeter determinations is failure to secure an *average* sample of steam. It is extremely doubtful whether such a sample is ever secured. To diminish the liability of error the instrument should be located as near as possible to the point where the sample is drawn off, and the sampling nipple should be placed as fully described in "Rules for Conducting Boiler Trials," Article XIV, page 204.

Taking an Observation—Locate the sampling nipple as above directed, attach the instrument as close to it as possible, and cover all exposed parts to prevent radiation. If the throttling calorimeter be used, locate the steam gauge on the pressure side, and the thermometer on the expansion side. To take an observation, note simultaneously the gauge reading and the thermometer reading, and from these the content of moisture may be determined directly from Fig. 15 or by use of formula [7]. If the separating calorimeter be used, attach to the separator outlet a piece of hose which terminates in a vessel of water on a platform scale graduated to read to $\frac{1}{100}$ of a pound. Similarly connect the steam outlet to another vessel of water resting on an equally sensitive scale. Note in each case the weight of each vessel including the water it contains. When ready to take an observation, blow out the instrument thoroughly, so there will be no water in the separator. Then simultaneously close the separator drip and insert the steam hose into its vessel of water. When the separator has accumulated a sufficient quantity of water, close the valve at the main steam pipe, thus cutting off the supply of steam to the instrument, remove the steam hose from the vessel of water into which it was inserted, and empty the separator water into its vessel on the scale. Note the final weight of each vessel and contents, then the differences between final and original weights will be respectively, the weight of moisture collected by the separator, and the weight of steam from which this moisture was taken, hence the proportion of moisture can be computed from formula [9].

Before taking any calorimeter observations, steam should be allowed to flow through freely until the instrument is thoroughly heated up.

5,400 H. P. OF STIRLING BOILERS, MONONGAHELA LIGHT & POWER CO., RANKIN, PA.

Flow of Steam Through Pipes and Orifices

Formulas for the flow of steam through pipes are based upon Bernoulli's theorem for the flow of water through circular pipes with friction, modified by inserting the proper constants for steam. The loss of energy due to friction is given by Unwin (from Weisbach) as

$$E_f = \frac{f2v^2WL}{gd} \qquad [10]$$

where E is the energy loss in foot pounds, due to the friction of W units (of weight) of steam passing through a pipe d feet in diameter and L feet long, with a velocity of v feet per second; g represents the acceleration of gravity (32.2) and f the coefficient of friction, which varies with the velocity to a certain extent, and with the size of the pipe. Some authorities consider both of these considerations negligible and treat f as a constant. In this article it will be regarded as varying according to the size of the pipe only, that is,

$$f = K\left(1 + \frac{3}{10d}\right) \qquad [11]$$

which relation was established by Unwin for a velocity of 100 feet per second. K is a constant experimentally determined and d the diameter of the pipe in feet.

If h be the loss of head in feet, then

$$E_f = Wh = \frac{f2v^2WL}{gd} \qquad [12]$$

$$\therefore h = \frac{f2v^2L}{gd} \qquad [13]$$

If D represents the density or weight of steam per cubic foot, and p the loss of pressure in pounds per square inch, due to friction, then

$$p = \frac{hD}{144} \qquad [14]$$

and from [11], [13] and [14],

$$p = \frac{Dv^2L}{72gd} \times K\left(1 + \frac{3}{10d}\right) \qquad [15]$$

Let d_1 = diameter of pipe in inches = $12d$. Let w = the flow in pounds per minute, then $w = 60v \times \frac{\pi}{4}\left\{\frac{d_1}{12}\right\}^2 D$, hence $v = \frac{9.6w}{\pi d^2 D}$ which when substituted in [15] gives

$$p = 0.04839\, K\left\{1 + \frac{3.6}{d_1}\right\}\frac{w^2L}{Dd_1^5} \qquad [16]$$

The following experimental determinations of K have been made:

K = .005 for water. (Unwin)
= .005 for air. (Arson)
= .0028 for air. (St. Gothard Tunnel Exp.)
= .0026 for steam. (Carpenter, Oriskany)
= .0027 for steam. (G. H. Babcock)

Using the value K = .0027, and substituting in [16] gives

$$p = 0.000131\left\{1 + \frac{3.6}{d_1}\right\}\frac{w^2L}{Dd_1^5} \qquad [17]$$

$$\text{Hence } w = 87\left\{\frac{pDd_1^5}{\left\{1 + \frac{3.6}{d_1}\right\}L}\right\}^{\frac{1}{2}} \qquad [18]$$

TABLE 19

FLOW OF STEAM THROUGH PIPES

Initial Gauge Pressure, Pounds per Square Inch.	DIAMETER* OF PIPE IN INCHES. LENGTH OF PIPE = 240 DIAMETERS.													
	¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
	WEIGHT OF STEAM PER MINUTE, IN POUNDS, WITH ONE POUND LOSS OF PRESSURE.													
1	1.16	2.07	5.7	10.27	15.45	25.38	46.85	77.3	115.9	211.4	341.1	502.4	804	1177
10	1.44	2.57	7.1	12.72	19.15	31.45	58.05	95.8	143.6	262.0	422.7	622.5	996	1458
20	1.70	3.02	8.3	14.94	22.40	36.04	68.20	112.6	168.7	307.8	496.5	731.3	1170	1713
30	1.91	3.40	9.4	16.84	25.35	41.63	76.04	126.9	190.1	346.8	559.5	824.1	1318	1930
40	2.10	3.74	10.3	18.51	27.87	45.77	84.49	139.5	209.0	381.3	615.3	906.0	1450	2122
50	2.27	4.04	11.2	20.01	30.13	49.48	91.34	150.8	226.0	412.2	665.0	979.5	1567	2294
60	2.43	4.32	11.9	21.38	32.19	52.87	97.60	161.1	241.5	440.5	710.6	1046.7	1675	2451
70	2.57	4.58	12.6	22.65	34.10	56.00	103.37	170.7	255.8	466.5	752.7	1108.5	1774	2596
80	2.71	4.82	13.3	23.82	35.87	58.91	108.74	179.5	269.0	490.7	791.7	1166.1	1866	2731
90	2.83	5.04	13.9	24.92	37.52	61.62	113.74	187.8	281.4	513.3	828.1	1219.8	1951	2856
100	2.95	5.25	14.5	25.96	39.07	64.18	118.47	105.6	293.1	534.6	862.6	1270.1	2032	2975
120	3.16	5.63	15.5	27.85	41.93	68.87	127.12	209.9	314.5	573.7	925.6	1363.3	2181	3193
150	3.45	6.14	17.0	30.37	45.72	75.09	138.61	228.8	343.0	625.5	1009.2	1486.5	2378	3481

*Diameters up to 5 inches inclusive are *actual* internal diameters of standard pipe, per Table 61, p. 213.

500 H. P. OF STIRLING BOILERS, HONOLULU BREWING & MALTING CO., LT'D., HONOLULU, H. I.

in which w=the flow in pounds per minute.
p=difference in pressure between the two ends of the pipe, in pounds per square inch.
D=density, or weight, per cubic foot of steam.
d_1=diameter of pipe in inches.
L=length of pipe in feet.

Table 19 is based on formula [18] and gives approximately the weight of steam per minute which will flow from various initial pressures, with one pound loss of pressure, through straight, smooth pipes, each having a length of 240 diameters.

For any assumed pipe length and loss, the weight will be

$$Q_1 = Q\left\{\frac{240dl}{\cdot\cdot}\right\}^{\frac{1}{2}} \qquad [21]$$

Example: Find the weight of steam of 100 lbs. initial gauge pressure which will pass through a 6″ pipe 720 feet long with a drop of 4 lbs. Under the conditions in the Table, 293.1 lbs. will pass, hence $Q = 293.1$

and $Q_1 = 293.1\left\{\frac{240 \times 6 \times 4}{720 \times 12}\right\}^{\frac{1}{2}} = 239.3$ lbs

Table 20 is due to Mr. E. C. Sickles, who used formula [16] with Prof. Carpenter's

TABLE 20

FLOW OF STEAM THROUGH PIPES

LENGTH OF PIPE ONE THOUSAND FEET

DISCHARGE IN POUNDS PER MINUTE CORRESPONDING TO DROP IN PRESSURE ON RIGHT FOR PIPE DIAMETERS IN INCHES IN TOP LINE.											DROP IN PRESSURE IN POUNDS PER SQUARE INCH CORRESPONDING TO DISCHARGE ON LEFT; DENSITIES AND CORRESPONDING ABSOLUTE PRESSURES PER SQUARE INCH IN FIRST TWO LINES.								
Diameter.	12″	10″	8″	6″	4″	3″	2½″	2″	1½″	1″	Density.	.208	.230	.284	.328	.401	.443	.506	.548
											Pressure.	90	100	125	150	180	200	230	250
Discharge	2328	1443	799	371	123.	55.9	28.8	18.1	6.81	2.52	Drop	18.10	16.4	13.3	11.1	9.39	8.50	7.44	6.87
"	2165	1341	742	344	114.6	51.9	27.6	16.8	6.52	2.34	"	15.60	14.1	11.4	9.60	8.09	7.33	6.41	5.92
"	1996	1237	685	318	106.	47.9	26.4	15.5	6.24	2.16	"	13.3	12.0	9.74	8.18	6.90	6.24	5.47	5.05
"	1830	1134	628	292	97.0	43.9	25.2	14.2	5.95	1.98	"	11.1	10.0	8.13	6.83	5.76	5.21	4.56	4.21
"	1663	1031	571	265	88.2	39.9	24.0	12.9	5.67	1.80	"	9.25	8.36	6.78	5.69	4.80	4.34	3.80	3.51
"	1580	979	542	252	83.8	37.9	22.8	12.3	5.29	1.71	"	8.33	7.53	6.10	5.13	4.32	3.91	3.42	3.16
"	1497	928	514	239	79.4	35.9	21.6	11.6	5.00	1.62	"	7.48	6.76	5.48	4.60	3.88	3.51	3.07	2.84
"	1414	876	485	226	75.	33.9	20.4	10.9	4.72	1.53	"	6.67	6.03	4.88	4.10	3.46	3.13	2.74	2.53
"	1331	825	457	212	70.6	31.9	19.2	10.3	4.43	1.44	"	5.91	5.35	4.33	3.64	3.07	2.78	2.43	2.24
"	1248	873	428	199	66.2	23.9	18.0	9.68	4.15	1.35	"	5.19	4.69	3.80	3.19	2.60	2.44	2.13	1.97
"	1164	722	400	186	61.7	27.0	16.8	9.03	3.86	1.26	"	4.52	4.09	3.31	2.78	2.34	2.12	1.86	1.72
"	1081	670	371	172	57.3	25.9	15.6	8.38	3.68	1.17	"	3.90	3.53	2.86	2.40	2.02	1.83	1.60	1.48
"	908	619	343	159	52.9	23.9	14.4	7.74	3.40	1.08	"	3.32	3.00	2.43	2.04	1.72	1.56	1.36	1.26
"	915	567	314	146	48.5	21.9	13.2	7.10	3.11	0.99	"	2.79	2.52	2.04	1.72	1.45	1.31	1.15	1.06
"	832	516	286	132	44.1	20.0	12.0	6.45	2.83	0.90	"	2.31	2.09	1.69	1.42	1.20	1.08	.949	.877
"	748	464	257	119	39.7	18.0	10.8	5.81	2.55	0.81	"	1.87	1.69	1.37	1.15	.97	.878	.769	.710
"	665	412	228	106	35.3	16.0	9.6	5.16	2.20	0.72	"	1.47	1.33	1.08	.905	.762	.690	.604	.558
"	582	361	200	92.8	30.9	14.0	8.4	4.52	1.98	0.63	"	1.13	1.02	.828	.695	.586	.531	.456	.429

To get the pressure drop for lengths other than 1,000 feet, multiply by lengths in feet ÷ 1,000.

To apply the table when the pipe lengths and the loss in pressure differ from those assumed, let L = the length, and d=the diameter of pipe, both in *inches*; l = the loss in pounds; Q = the weights as given in the table, and Q_1 = the weight under the changed conditions, then:

For any length of pipe, if the weight of steam passing is the same as given in the table, the loss will be

$$l = \frac{L}{240d} \qquad [19]$$

If the pipe length is the same as assumed in the table, but the loss is different, then the quantity passing will be

$$Q_1 = Ql^{\frac{1}{2}} \qquad [20]$$

value K = 0.0026. To use the table, assume a certain drop in pressure. Look for this drop in the column at the right under the heading "Drop in pressure in pounds;" next pass to the left along a horizontal line, until under heading "Discharge in pounds per minute" the tabular quantity which corresponds nearest to the quantity desired, is found; the size of pipe given at the top of the column in which the tabular quantity is located will be the one required.

Elbows, globe valves, and a square ended entrance to the pipe, all offer resistance to the passage of the steam; it is convenient to consider this resistance equivalent to a length of straight pipe, and add these equivalent lengths to the straight portions of

the pipe line to obtain the total length to be used in the formulas. Complicated formulas for determining the equivalent length have been worked out, but in view of the varying proportions of valves and fittings such formulas are not worth the time it takes to apply them, and for all practical purposes it will be sufficiently accurate to allow for resistance at the entrance of a pipe a length equal to 60 times the diameter; for a right angled elbow a length of 40 diameters, and for a globe valve a length equal to 60 diameters.

the pipe to 6,000 ft. per minute. When the pipes are long, this sometimes gives a greater drop in pressure than is desirable, and it is then best to check the sizes by referring to the tables.

In marine work, a velocity of 9,000 ft. per minute in steam pipes is very often used with excellent results, and there is no reason why this cannot be done in stationary practise, provided the boilers can be worked at a pressure sufficient to compensate for the drop in the pipe line. See the chapter on Steam Piping, page 213.

TABLE 21

FLOW OF STEAM INTO THE ATMOSPHERE

Absolute Initial Pressure per Square Inch. Pounds.	Velocity of Outflow at Constant Density, Feet per Second.*	Actual Velocity of Outflow, Expanded. Feet per Second.	Discharge per Square Inch of Orifice per Minute. Pounds.	Horse-Power per Square Inch of Orifice if H.P.= 30 lbs. per Hour.
25.37	863	1,401	22.81	45.6
30.	867	1,408	26.84	53.7
40.	874	1,419	35.18	70.4
50.	880	1,429	44.06	88.1
60.	885	1,437	52.59	105.2
70.	889	1,444	61.07	122.1
75.	891	1,447	65.30	130.6
90.	895	1,454	77.94	155.9
100.	898	1,459	86.34	172.7
115.	902	1,466	98.76	197.5
135.	906	1,472	115.61	231.2
155.	910	1,478	132.21	264.4
165.	912	1,481	140.46	280.9
215.	919	1,493	181.58	363.2

Drop in pressure in a steam pipe does not necessarily indicate a loss of energy, because the friction which causes the drop transforms the energy into heat, and this evaporates moisture and superheats the steam. The superheating effect is very slight ordinarily, but will be very manifest if the pressure drop is large, as illustrated in the throttling calorimeter.

A common rule in laying out piping is to limit the velocity of the steam through

Flow of Steam into the Atmosphere—When steam is discharged into the atmosphere, the velocity of outflow (at constant density and when the absolute pressures are greater than 1.73 times the atmospheric pressure) is as given in Table 21.

The external pressure per square inch has been taken as that existing under the standard atmospheric pressure of 14.7 lbs. absolute—while the ratio of expansion in the nozzle itself has been taken as 1.624.

**I. e.*, if the steam maintained the same density as it had at the initial pressure.

Napier's approximate formula for the outflow of steam into the atmosphere is

$$\text{Pounds of steam per second} = \frac{pa}{70} \qquad [22]$$

In which p = absolute pressure in pounds per square inch, and a = area of orifice in square inches. This formula gives results which correspond very closely with those in Table 21 as shown below:

p	Discharge, Pounds, per Minute.	
	By Table 21.	By Napier's Rule.
25.37	22.81	21.74
40.	35.18	34.29
60.	52.59	51.43
75.	65.30	64.29
100.	86.34	85.71
135.	115.61	115.71
165.	140.46	141.43
215.	181.58	184.29

Prof. Peabody conducted a series of experiments on flow of steam through tubes ¼-inch in diameter and ¼-inch to ½-inch, and 1½-inch long, with rounded entrances, in which the results agreed closely with Napier's formula, the greatest difference being an excess of the experimental over the calculated result, of 3.2 per cent.

Flow of Steam from Orifices into a Pressure Above that of the Atmosphere—The flow of steam of a higher towards a lower pressure increases as the difference of pressure is increased, until the external pressure becomes only 58 per cent. of the absolute initial pressure. Below this point, the flow of steam is neither increased nor diminished by a reduction of external pressure, even to the extent of a perfect vacuum. Table 22, selected from Mr. Brownlee's data, illustrates this fact.

The following formula is frequently used to determine the flow of steam through an orifice against a pressure greater than two-thirds the discharge:

$$W = 1.9\, AK(P-p)^{\frac{1}{2}}\, p \qquad [23]$$

where W = weight of escaping steam in pounds per minute.

A = area of orifice in square inches.

K = 0.93 for a short pipe and 0.63 for an opening such as a hole in a plate or a safety valve.

P = absolute pressure of steam, pounds per square inch.

p = difference in pressure between the two sides in lbs. per square inch.

TABLE 22

FLOW OF STEAM THROUGH ORIFICES (*Brownlee*)

Absolute Pressure in Boiler per Sq. In. Pounds.	Absolute External Pressure per Square Inch. Pounds.	Ratio of Expansion in Nozzle.	Velocity of Outflow at Constant Density. Feet per Second.	Actual Velocity of Outflow Expanded. Feet per Second.	Discharge per Square Inch of Orifice per Min. Pounds.
75	74	1.012	227.5	230.	16.68
75	72	1.037	386.7	401.	28.35
75	70	1.063	490.	521.	35.93
75	65	1.136	660.	749.	48.38
75	61.62	1.198	736.	876.	53.97
75	60	1.219	765.	933.	56.12
75	50	1.434	873.	1252.	64.
75	45	1.575	890.	1401.	65.24
75	43.46	1.624	890.6	1446.5	65.3
75	15	1.624	890.6	1446.5	65.3
75	0	1.624	890.6	1446.5	65.3

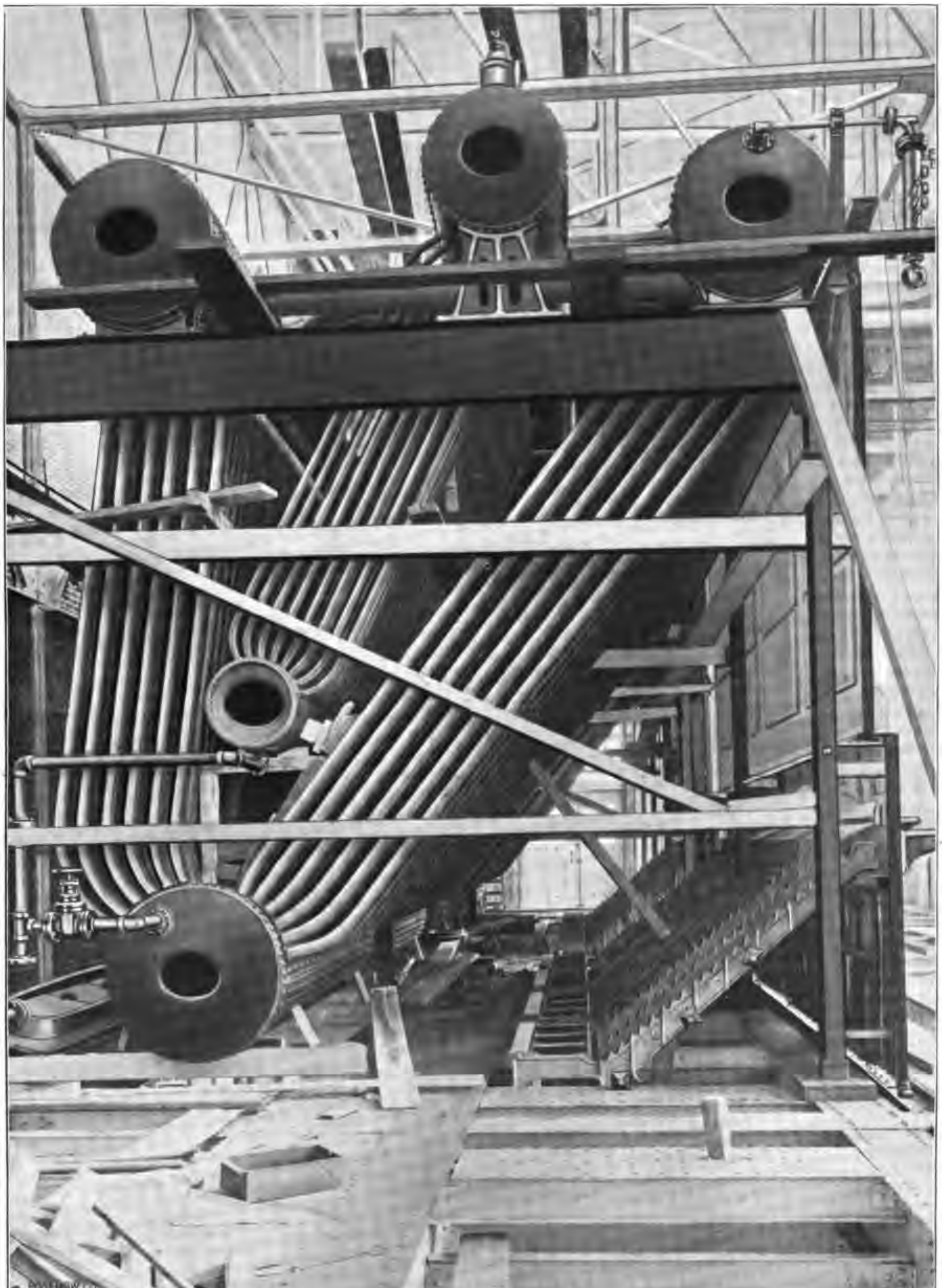


FIG. 18. THE STIRLING SUPERHEATER BOILER AS INSTALLED FOR THE GENERAL ELECTRIC CO.
16,000 H. P. OF STIRLING BOILERS OPERATED BY THIS COMPANY

Superheated Steam and the Stirling Superheater

Superheated steam is steam whose temperature exceeds that of saturated steam of the same pressure, and it is produced by adding additional heat to saturated steam which has been removed from contact with the water from which it was formed. Its properties approximate those of a perfect gas, and its thermal conductivity is lower than that of saturated steam.

Superheated steam is used because:

(1) There is always a loss of heat by radiation from steam pipes, and the heat so lost represents an equivalent condensation when the pipe conveys saturated steam. Superheated steam cannot condense; it must first lose all of its superheat and be reduced to saturated steam. In consequence, if sufficiently superheated it can lose the amount of heat represented by radiation from the steam pipes, yet reach the engine perfectly dry. Since the thermal conductivity of superheated steam is less than that of saturated steam, the heat will not be so rapidly transmitted from the body of the steam to the walls of the pipe.

(2) In an engine the steam is admitted into a space which has been cooled by the steam exhausted during the previous stroke. The heat necessary to warm the cylinder walls from the exhaust temperature to the temperature of the entering steam can be supplied only by the entering steam, hence if it be saturated some of it must condense. The amount thus condensed is seldom less than 20 to 30 per cent. of the total weight of steam entering the cylinder. It is obvious, however, that if an amount of heat more than sufficient to warm the cylinder walls could, by means of superheating, be imparted to the steam before it reached the engine, then even after the cylinder walls had been warmed up the steam would remain dry, and the initial condensation would thus be overcome.

These properties of superheated steam have long been known, but their practical application has been slow, owing to constructive difficulties. The recent successful development of the steam turbine, and of reciprocating engines adapted to the use of superheated steam, has rendered necessary the development of a simple, durable, efficient and safe steam superheater. The Stirling Company therefore inaugurated an exhaustive series of researches and experiments on superheated steam, and these have resulted in the development of a superheater which has produced a higher degree of superheat than has yet been recorded as obtained from any other type of superheater installed in connection with a boiler. In addition to this, its constructive features are as radical an improvement over previous superheaters as the Stirling boiler is over the types of boiler which preceeded it.

Before describing the Stirling superheater the principles governing the amount of superheating surface, and of the heating surface of the boiler to which the superheater is attached, will be explained and illustrated.

Specific Heat of Superheated Steam—The amount of fuel required to superheat steam, and the quantity of fuel that must be burned to produce this heat, are greater than is commonly supposed. The specific heat of superheated steam at atmospheric pressure and near the point of saturation was found by Regnault to be 0.48, and for the succeeding 50 years it was thought that this value of the specific heat applied to higher pressures. Recent investigations both in this country and in Europe have shown that the specific heat is not constant, and that it is approximately 0.65 for 100° superheat, and 0.75 for 200° superheat. Using these values it can be calculated that the fuel used to generate saturated steam must be increased by about the following percentages in order to superheat the steam to the degrees named.

TABLE 23

FUEL NEEDED FOR SUPERHEATING

DEGREE OF SUPERHEAT	ADDITIONAL FUEL NEEDED
75°	5%
100°	7
150°	11
200°	15
250°	20

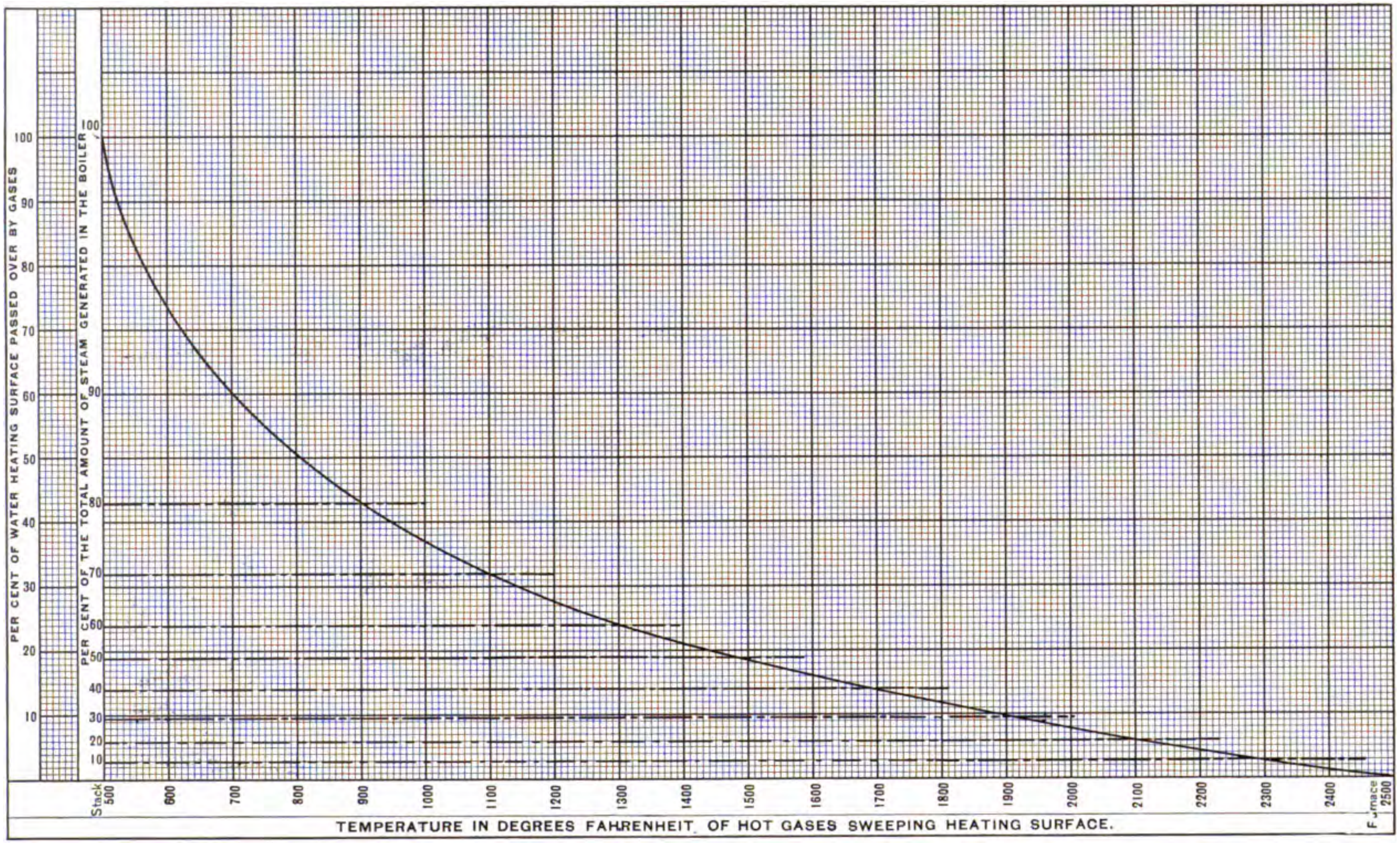


FIG. 19. CURVE SHOWING RELATION BETWEEN GAS TEMPERATURE, HEATING SURFACE PASSED OVER, AND AMOUNT OF STEAM GENERATED
TEN SQUARE FEET OF HEATING SURFACE ARE ASSUMED AS EQUIVALENT TO ONE BOILER HORSE-POWER

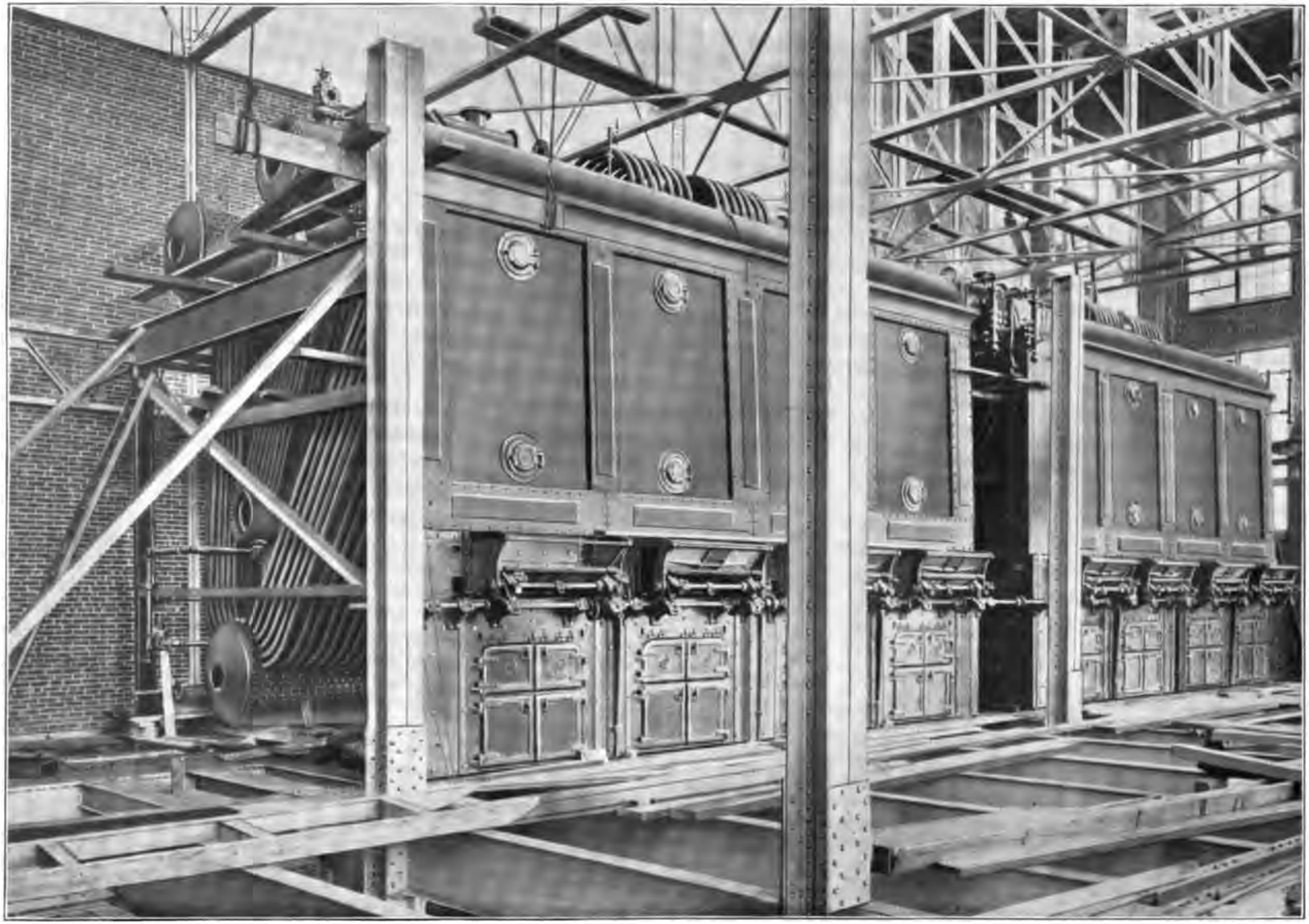
The degree of superheat being assumed, the amount of superheating surface required to produce it will depend upon where the surface is located in the path of the hot gases, between the furnace and breeching. This principle is of the utmost importance in superheater design, hence it will be further illustrated by means of the curve in Fig 19. In this the abscissas represent the temperature of the hot gases at different points in their path from the boiler furnace to the breeching; the column on the extreme left indicates the per cent. of boiler heating surface passed over by the gases, and the adjoining column gives the amount of steam generated by the heat absorbed from the gases, this amount of steam being expressed as a per cent. of the total steam generated in the boiler. Example: When the gases are cooled to 700°, they have passed over 60% of the boiler heating surface, and the heat they have given up has generated 90% of the steam which the boiler is producing.

In drawing the curve, 10 square feet of heating surface have been taken as the equivalent of one boiler horse-power*, in conformity with the usual practise of builders of water-tube boilers. The furnace temperature has been assumed as 2500° F., as the result of many experiments made by this Company's engineers. The breeching temperature of 500°. is assumed as a fair average of conditions for best economy. This temperature may appear to be too high, in view of the statement current in engineering literature, that for each reduction of 100° in the breeching temperature the boiler efficiency is increased 6%. In order that this statement may be true the air supply per pound of fuel, the degree of completeness of the combustion, the losses by radiation and air leakages, the kind of fuel, and other factors, must remain unchanged—a requirement which cannot be met in practise. A deficient air supply will cause the volatile matter in the fuel to pass off unburnt, and while this may lower the breeching temperature, it lowers the efficiency also. In consequence of these facts, and as further proved by many tests, a low breeching temperature does not necessarily indicate high efficiency, and an increase in the temperature often augments the efficiency, hence the temperature of 500° assumed in computing the curve is very nearly that which gives the best results under average conditions.

The curve connecting the furnace and breeching temperatures was plotted from an equation based upon the assumption that the heat transferred from the gases to the water is directly proportional to the difference in temperature; that is, if the temperature difference is 1000°, each square foot of surface will absorb twice as much heat as it would with a temperature difference of 500°. This was the original assumption of Rankine, and while its accuracy was later questioned, many hundred temperature measurements at different points along the path of the gases in Stirling boilers conform more closely to this curve than to any other. The liability of error is greatest at the lower portions of the curve where the temperatures are highest, because at these points large quantities of heat are transmitted directly from the glowing coals to the heating surface by radiation, while farther along the path of the gases the heat is transmitted by convection. This possible source of error, does not, however, effect either the part of the curve to be used in the following discussion, or the general conclusions to be deduced from it.

In their path through the boilers the gases drop from 2500° to 500°, a difference of 2000°. If equal drops in temperature represented equal amounts of heat given out and absorbed by the boiler, then each 200° drop in temperature would represent 10% of the total heat absorbed by the boiler. This is not literally true, however, because the specific heat of the gases is greater at high than at low temperatures, but the difference is not sufficient to affect the important conclusions to be drawn from the curve. Accordingly, to determine the figures in the second column on the left side of Fig. 19, horizontal lines have been drawn for each 200° drop in the gas temperatures, and the corresponding per cent. of the total steam generated, up to the point where each drop is noted, is written in the column. For example, when the gas temperature has dropped to 1500°, then 19% of the heating surface has been passed over, and 50% of the total output of steam has been generated by this 19% of heating surface. If the gases were allowed to escape at this point instead of continuing

*See definition, page 195.



PART OF 14,700 H. P. OF STIRLING SUPERHEATER BOILERS INSTALLED BY DETROIT EDISON COMPANY, DETROIT, MICH.

through the boiler, then 40% of the total available heat would have been utilized, and the heating surface per boiler horse-power would be 3.8 square feet; here and in the following part of this article the term horse-power refers to the horse-power of the saturated steam boiler, or the saturated steam portion of the superheater boiler, without reference to the additional capacity represented by the superheater.

Similarly the results for other drops in temperature can be calculated as follows:

TABLE 24

GAS TEMPERATURE.	HEATING SURFACE PASSED OVER.	STEAM GENERATED.	HEATING SURFACE PER H. P.	EFFICIENCY OF BOILER.
1500°	19%	50%	3. 8 sq. ft.	40%
1000°	37	75	4. 9 "	60
750°	54	87.5	6.17 "	70
500°	100	100	10.00 "	80

The efficiency of 80% is possible with large boilers burning high grade coal, oil, or gas.

Application of the Curve to the Problem of Superheater Design—A superheater may be either independently fired, or be placed in the setting of a boiler and absorb heat from the furnace gases which sweep over its surface on their way through the boiler. The latter form of superheater is the one most generally used, hence the principles underlying the design of this form of superheater will now be explained.

In order that the superheater boiler may develop the same thermal efficiency as the standard boiler used for generating saturated steam, the furnace temperature, the breeching temperature, and the weight of flue-gases per pound of fuel, must be the same for either type of boiler. Assume that the superheater is to be located in the rear of the boiler, and that 100° of superheat will be required. From Table 23 this degree of superheat will require 7% more fuel than is required to generate an equal weight of steam. Referring to the curve, Fig. 19, 7% of the total heat absorbed represents a drop of temperature of 140°, therefore, as the breeching temperature is to remain unchanged, the gases must enter the superheater at a temperature of 140°+500°=640°. Locating this temperature on the curve, it is found to correspond to a point where 68% of the heating surface of the boiler has been passed over by the furnace gases. Consequently 32% of the boiler heating surface of the standard boiler must be replaced by superheating surface sufficient to absorb 7% of the total heat absorbed by the boiler. The effect of this substitution will then evidently be a reduction of 7% in the weight of saturated steam generated, and a reduction of 32% in the heating surface of the boiler, so that 68% of the heating surface of the saturated steam boiler generates 93% of the weight of steam produced by that boiler, and the heating surface per boiler horse-power will be $\frac{6.8}{.93}$=7.13 square feet. It is evident that if more boiler heating surface per horse-power be installed, the gases will be cooled to a temperature below 640° before entering the superheater, in which case the required degree of superheat will not be obtained, hence it at once follows that the boiler heating surface of a superheater boiler must be proportioned in a different manner from that in a saturated steam boiler, as will more clearly be developed later on.

Since the purpose of this investigation is to determine the relation between superheating surface, and the heating surface of the saturated steam portion of the boiler to which the superheater is connected, it is to be understood that in the remainder of this chapter the term "boiler heating surface" denotes the heating surface of that part of the combination of boiler and superheater which generates saturated steam, while the term "superheating surface" refers to the surface which superheats that steam after it is generated.

If 200° of superheat be required, 15% of the total heat utilized must be absorbed by the superheater, which will correspond to a reduction of 300° in the gas temperature; this would require 50% of the heating surface of the saturated steam boiler to be replaced by superheating surface, and the remaining 50% of the boiler surface would have a capacity of 85% of that of the saturated steam boiler, hence would have $\frac{5}{.85}$ = 5.9 square feet of boiler heating surface per horsepower. For 75° superheat, 25% of the boiler

SUPERHEATER BOILERS, BURNING CRUDE OIL, EDISON ELECTRIC COMPANY, LOS ANGELES, CAL., 6,000 H. P. OF STIRLING BOILERS OPERATED BY THIS COMPANY

heating surface would be replaced by the superheater, and there would be 7.9 square feet of boiler heating surface.

Instead of locating the superheater behind the boiler it may be inserted at some intermediate point in the path of the gases. For instance, assume that the superheater replaces three-tenths of the heating surface of the standard boiler, and is so placed that four-tenths of the amount of heating surface of the standard boiler is located ahead of the superheater, and the other three-tenths is placed behind it. Then the part of the curve between 40% and 70% in the left hand column will represent the cooling of the gas while passing over the superheater; the drop of temperature will be 300°, which represents 15% of the total heat absorbed, hence from Table 23 the degree of superheat will be 200°. The boiler will produce 85% as much steam as the saturated steam boiler, and the boiler heating surface per horse-power is $\frac{7.0}{.85} = 8.24$ square feet.

If 100° superheat were required and the ratio of boiler heating surface in front of and behind the superheater be kept as in the preceding case, the steam production will be 7% less than in the saturated steam boiler, and the gas temperature will be reduced 140° in the superheater. The requirements can be met by substituting superheating surface in place of boiler heating surface between the points in the curve represented by 63% and 48% in the left hand column, and the boiler heating surface per horse-power will be $\frac{8.5}{.93} = 9.14$ square feet.

By removing the superheater farther forward, as for instance on that part of the curve represented between 21% and 32% in the left hand column, the steam production would be 15% less than in the saturated steam boiler, the reduction of gas temperature in the superheater will be 300°, the superheat will be 200°, and the boiler heating surface per horse-power will be $\frac{8.9}{.85} = 10.5$ square feet.

It will, however, be found that in almost the same proportion that the boiler heating surface per horse-power is decreased, the necessary superheating surface will increase, so that the sum of the boiler heating surface and superheating surface per boiler horse-power will be very nearly the same for any given degree of superheat.

From the preceding discussion it follows that if a saturated steam boiler and a superheater boiler are to be of identical fuel effiiciency the following laws will hold:

(1) *A superheater boiler must provide fewer square feet of boiler heating surface per horse-power than are required for a saturated steam boiler, provided the superheater is located at a point where at least 25% of the boiler heating surface is placed between the superheater and the furnace.*

(2) *The boiler heating surface per horse-power will be decreased as the per cent. of boiler heating surface in front of the superheater is increased.*

(3) *The position of the superheater remaining the same, the higher the superheat the less the boiler heating surface required per horse-power developed by the boiler heating surface.*

It therefore follows that the superheater may be placed either in the rear of all the boiler heating surface, or at some intermediate position with boiler heating surface ahead of and behind it. In the two cases the relative amount of boiler heating and superheating surface must vary if the results are to be the same.

The kind of engine operated by the steam has, however, a vital bearing upon the location of the superheater. The engine may be either a steam turbine, or a reciprocating engine whose working parts are so designed as to permit the use of superheated steam. For the steam turbine the degree of superheat is seldom less than 100°, and is usually higher, while the maximum superheat which may be used has yet to be determined. In a reciprocating engine the superheat which may be used to advantage is limited by the design of the working parts, and any considerable increase augments the difficulty of lubrication, and may cause trouble with the packings, etc., hence the superheater should be so located that when the boiler is forced the steam temperature will not exceed the limit which is safe for such an engine.

These requirements may be met by properly locating the superheater. If it be placed in the middle pass of the boiler the close

proximity of the superheating surface and the furnace will cause the degree of superheat to rise at times much faster than can occur when the superheater is placed in the rear pass. For this reason the superheater located in the middle pass is to be

Fig. 20 represents a vertical section of the boiler and superheater, as arranged for superheats not exceeding 100°. The superheater is located behind the boiler heating surface, and this design is particularly adapted to operating reciprocating engines.

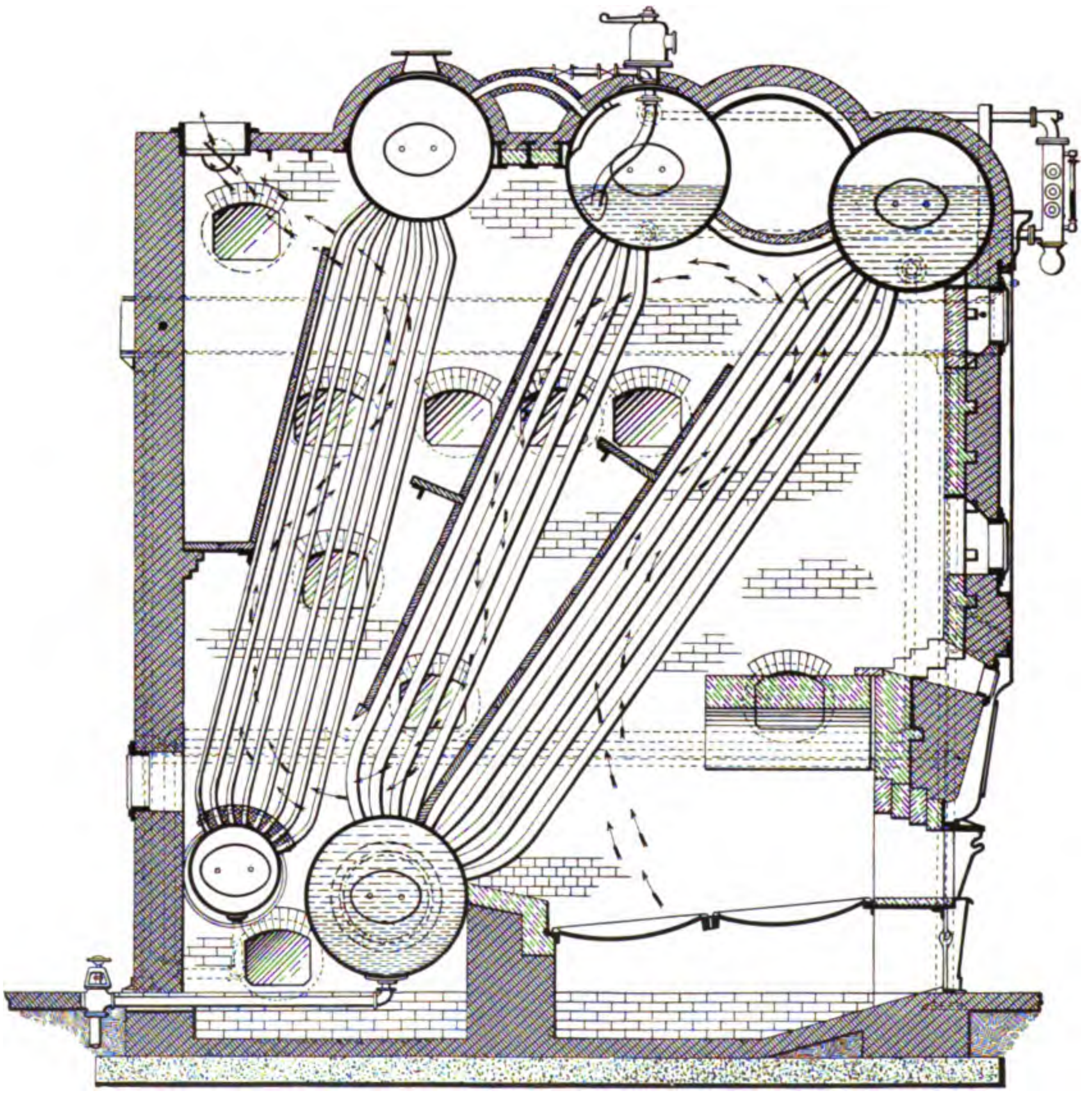


FIG. 20. SECTIONAL SIDE ELEVATION OF STIRLING BOILER WITH SUPERHEATER IN REAR PASS

preferred for operating steam turbines where the superheat exceeds 100°, while one located in the rear pass is most suitable for supplying steam to a reciprocating engine.

The Stirling Superheater Boiler is designed in conformity with these principles.

Fig. 21 represents the section used for degrees of superheat exceeding 100°. The superheater is placed between the two banks of boiler heating surface, and by properly proportioning the boiler heating and superheating surface, superheats up to 250°

can be obtained. In either case the superheater consists of two drums connected by seamless-drawn tubes two inches in diameter. The construction is identical with that of the standard design of Stirling boiler, and all of the advantages of the bent tube are retained. In addition the joint between the tube ends and the drums is protected from high heat by a layer of asbestos cement which rests upon the lower drum, or is supported on metal bars placed below the upper superheater drum.

Flooding the Superheater—When desired the superheater may be flooded, and used to generate saturated steam. Fig. 22 shows the arrangement of flooding pipe which connects the front steam drum with the lower superheater drum. When the superheater is used for generating saturated steam the three compartments in the upper drum are thrown into communication by suitable valves so that each compartment can discharge its quota of saturated steam into the main pipe.

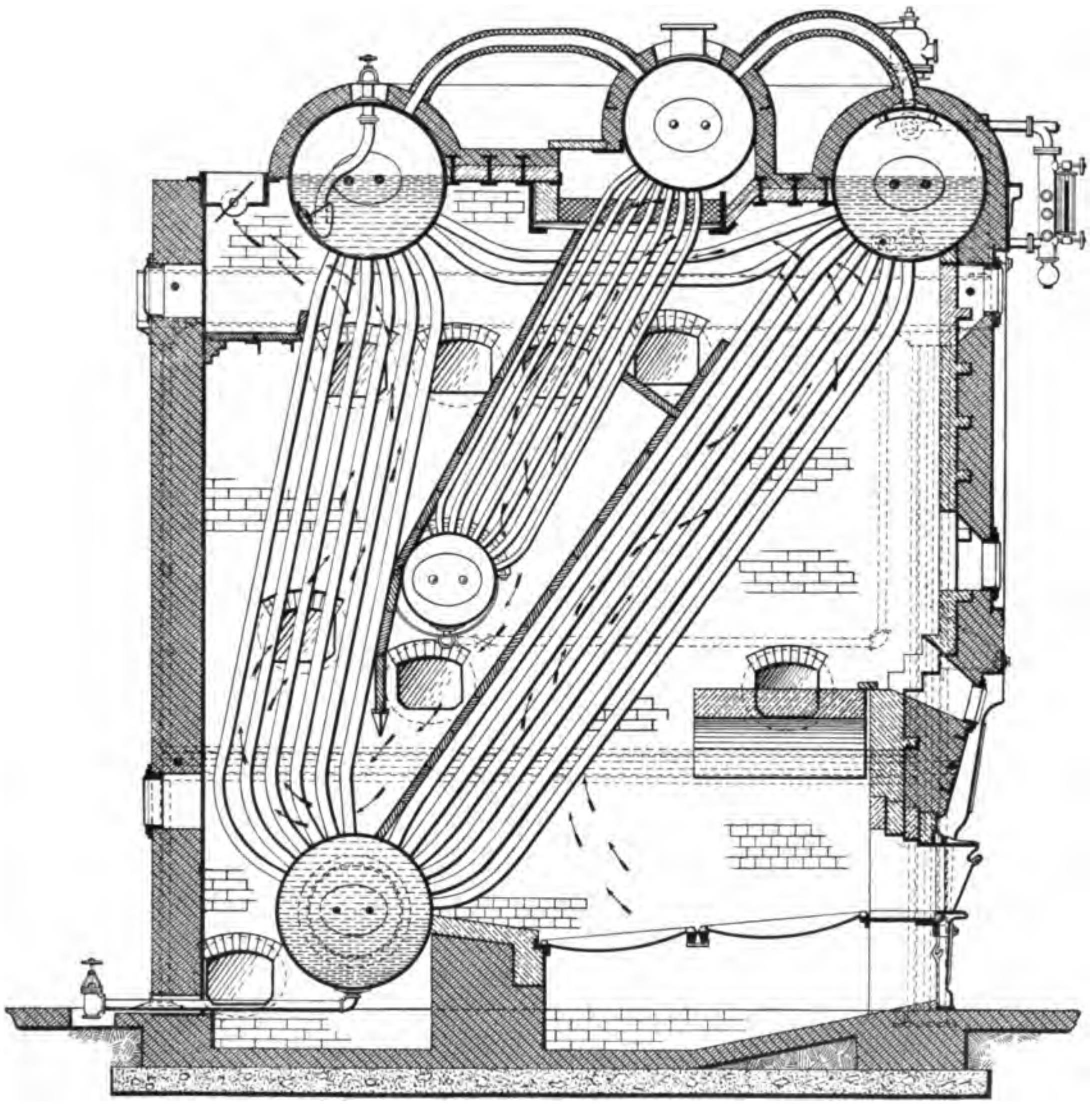


FIG. 21. SECTIONAL SIDE ELEVATION OF STIRLING BOILER WITH SUPERHEATER IN MIDDLE PASS

The advantages of this arrangement are obvious. The superheater boiler may, in emergencies, be used as a saturated steam boiler to operate other engines in the plant when those which use superheated steam are shut down. When a number of superheater boilers are operated together, and it is desired to reduce the degree of superheat, any one of these boilers may be used to generate saturated steam only, and this be mixed with the superheated steam to reduce the degree of superheat to any desired point.

Course of the Steam in the Superheater—The upper drum in the superheater is divided into three compartments by means of two partitions, and the lower drum is similarly divided into two compartments; each partition either contains a manhole, or is removable, so that all parts of the drum are accessible. The steam enters one end compartment of the upper drum, and makes four passes through the tubes, as indicated in Fig. 23.

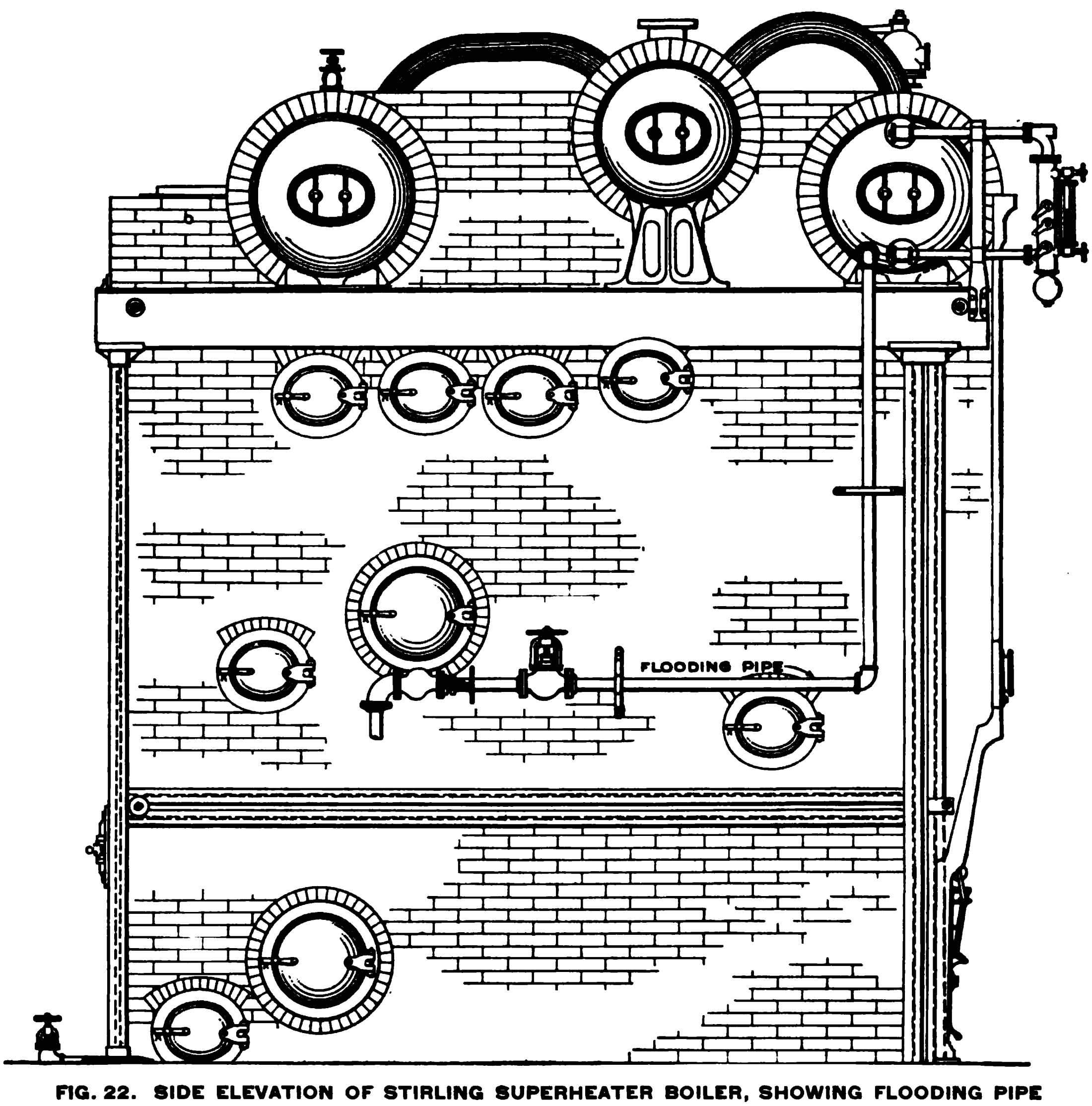


FIG. 22. SIDE ELEVATION OF STIRLING SUPERHEATER BOILER, SHOWING FLOODING PIPE

Independently Fired Superheater—Fig. 24 represents the Stirling Independently

Fired Superheater. In this all the constructive advantages of the Stirling boiler are retained. The saturated steam from the main boiler plant enters the rear superheater drum, passes through the rear bank of tubes into the lower drum, thence to the upper drum, from which it passes into the pipe line. The furnace is similar to that used in the standard design of Stirling boiler. To protect the superheater tubes from the high temperature of the furnace a sufficient amount of boiler heating surface is located in front of the superheater to

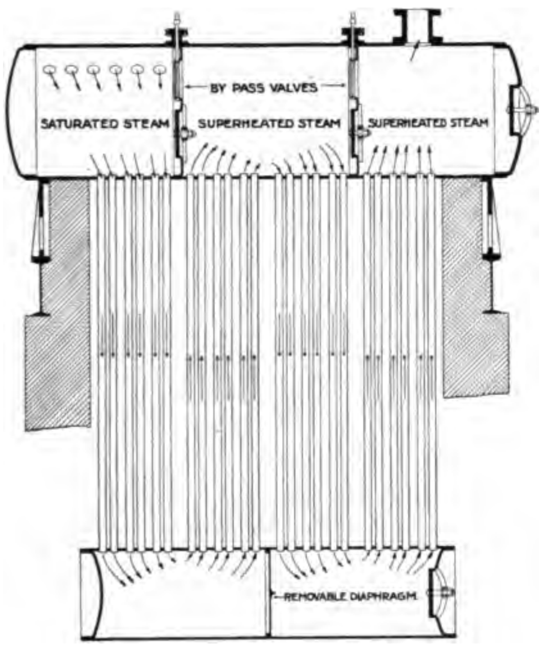


FIG. 23. SECTION THROUGH STIRLING SUPERHEATER, SHOWING PATH OF STEAM

reduce the temperature of the gases to 1500° by the time they reach the superheater tubes. Referring to the curve, Fig. 19, it will be noted that when the gas temperature reaches 1500° in the standard boiler, 19% of the boiler heating surface has been swept over by the gases, 50% of the steam produced by the boiler has been generated, and the boiler heating surface per horse-power is 3.8 square feet. Consequently in the independently fired superheater shown in Fig. 22 50% of the heat absorbed is used to generate steam which is added to the steam furnished by the main boiler plant, hence increases the capacity of the plant in proportion. The remaining 50% of the heat is absorbed by the superheater, and superheats both the steam from the main boiler plant and that from the front bank of water-tubes. If, for example, the degree of superheat is 150°, then from Table 23 it will take 11% as much heat to superheat a pound of steam as to generate it from water in form of saturated steam, hence for each pound of saturated steam generated in the front bank of the superheater, $9\frac{1}{11}$ pounds may be superheated, and $8\frac{1}{11}$ pounds are delivered from the boilers, therefore the superheater will generate about 12% of the amount of steam furnished by the main boiler plant.

As a further precaution against any possible overheating of the superheater tubes nearest to the furnace, a flap valve is placed in the pipe conveying saturated steam to the superheater, as shown in Fig. 24. The spindle of this valve is connected by links to the superheater damper, so that the damper opening is regulated according to the quantity of steam flowing into the superheater; if the steam flow stops, the valve drops to its seat, and the damper is closed.

To provide for circulation in the water-tubes, four "downcomer" tubes are placed at each end of the drum, as shown in the section on line A——B, Fig. 24. These are placed in a slot in the wall and are protected from heat by the tile as shown.

Independently Fired Superheaters can be furnished of any desired capacity, suitable for any superheat up to 250°.

Flooding Pipe—The upper water drum and lower superheater drum are connected by piping, as shown in Fig. 22, hence, if desired, the superheater sections may be flooded, converting the whole into a saturated steam boiler.

Removing Tubes from the Superheater Boiler—The 3¼-inch tubes in the boiler heating surface are alternately spaced 6¾ and 5¼ inches; the 2-inch superheater tubes are alternately spaced 4¼ and 3 inches. See Fig. 12.* This method of spacing permits any tube to be removed from the drums and be passed through the wider spacing, hence *any tube in the boiler or superheater can be replaced without disturbing other tubes.*

*Page 25.

Cleaning—An extremely important advantage is that the superheater tubes may be cleaned by means of a turbine cleaner in precisely the same manner as the regular boiler tubes are cleaned. *This point is of the utmost importance*, since when using feed waters containing vegetable matter, sewage, etc., some of this matter will be carried over into the superheater and deposited upon the tubes, hence no form of superheater that cannot be readily cleaned will meet the requirements of successful use, considered as a *superheater only*. When, as in case of the Stirling superheater, provision is made for flooding the superheater and using it in emergencies to generate saturated steam, it is evident that for satisfactory service it is necessary that the superheater tubes be just as readily cleaned as the water tubes. This requirement is perfectly met in the Stirling superheater.

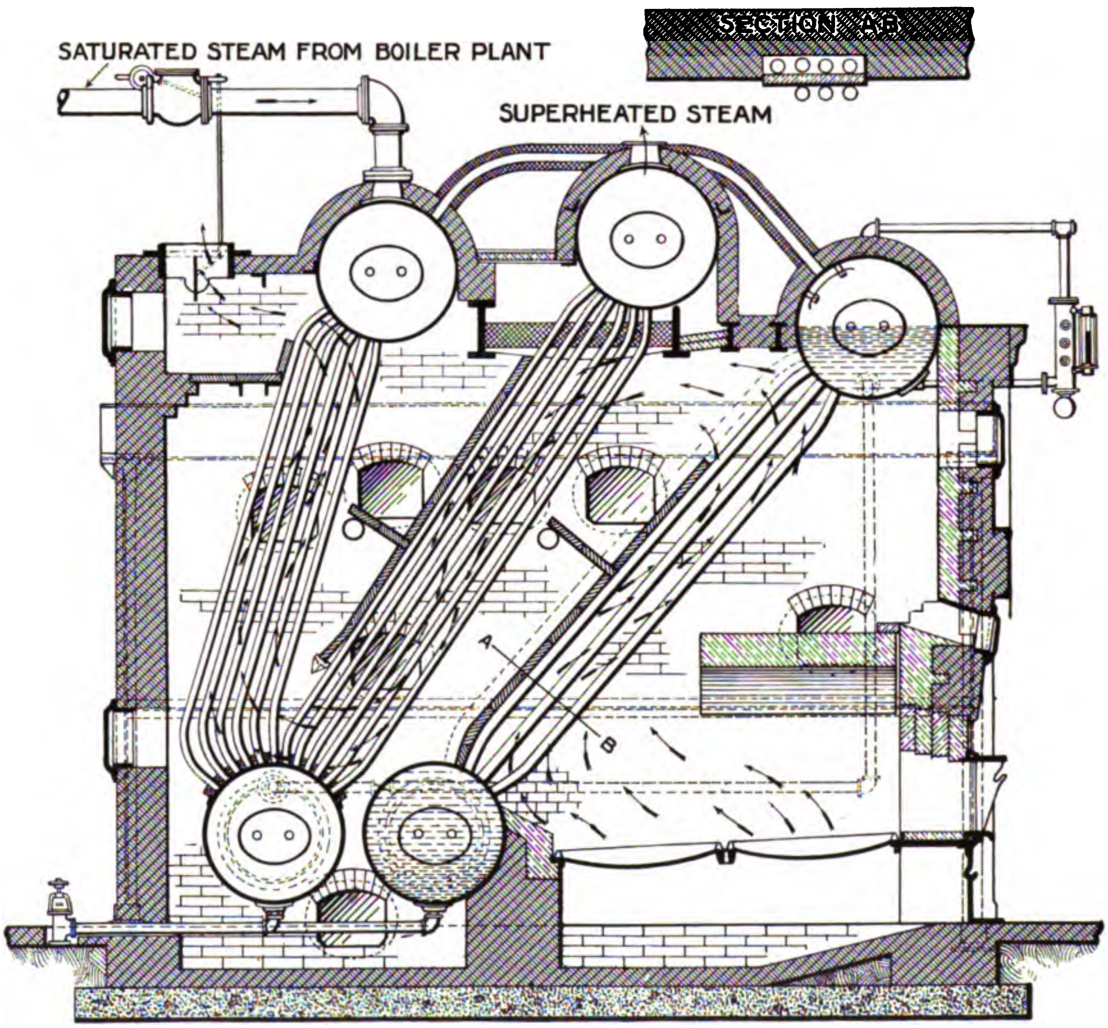


FIG. 24. SECTIONAL SIDE ELEVATION OF THE STIRLING INDEPENDENTLY FIRED SUPERHEATER

Superheaters for Boilers already Installed—The Stirling Company manufactures superheaters which may be attached to any Stirling boiler now in use, and will promptly furnish blue prints, prices, and full information on application from prospective purchasers.

Combustion

Combustion, as the term is used in steam engineering, is the rapid chemical combination of oxygen with carbon, hydrogen and sulphur, with the accompaniment of heat and light. The substance which combines with the oxygen is the *combustible.** The combustion is *perfect* when the combustible is oxidized to the highest possible degree; thus, conversion of carbon into carbon dioxide (CO_2) represents perfect combustion, while its conversion to monoxide (CO) is imperfect combustion, since the monoxide can be further burned and finally converted into CO_2.

Kindling Point—As in many other chemical processes, a certain degree of heat is necessary to cause the union of the oxygen and combustible; the temperatures necessary to cause this union are the kindling temperatures, and are approximately as given in the following table by Stromeyer.†

TABLE 25

KINDLING TEMPERATURES

Lignite Dust . . .	300° F.
Sulphur	470
Dried Peat . . .	435
Anthracite Dust .	570
Coal	600
Cokes	Red Heat
Anthracite . . .	" " 750
Carbon Monoxide .	" " 1211
Hydrogen . . .	1030 or 1290

The Oxygen necessary for combustion is supplied from the air. Its density is 1.10521, (Air = 1); its weight 0.088843 lbs. per cu. ft. at 32° F., and atmospheric pressure; its atomic weight is 16; a pound of air contains 0.2315 lbs. of oxygen, and one pound of oxygen is contained in 4.32 lbs. or air.

Carbon (C), the most abundent combustible, has atomic weight of 12, and reaches the boiler furnace as a constituent of oil, gas, coal, charcoal, wood, etc.

Hydrogen (H) occurs free in small quantity in some fuels, but is usually in combination with the carbon. Its atomic weight is 1; its density is 0.0692, (Air=1); and its weight per cubic foot at 32° F. and atmospheric pressure is 0.00559 lbs.

Sulphur (S, atomic weight 32) is found in most coals and in some oils. It is usually present in a combined form, either as sulphide of iron, or sulphate of lime; in the latter form it has no heating value. Its presence in fuel is objectionable because the gases formed from its combustion attack the metal of the boiler and causes rapid corrosion, particularly in presence of moisture.

Nitrogen (N) is drawn into the furnace with the air. Its atomic weight is 14; its density is 0.9701, (Air=1); its weight per cubic foot at 32° F. and atmospheric pressure is .07831 lbs.; each pound of air at atmospheric pressure contains 0.7685 lbs. of nitrogen, and one pound of nitrogen is contained in 1.301 lbs. of air.

Nitrogen performs no useful office in combustion, and passes through the furnace without change. It dilutes the air, absorbs heat and reduces the temperature of the products of combustion and is the chief source of heat loss in furnaces.

Combining Weights—When chemical elements unite to form a new compound they do so in definite proportions which are always the same, and the union produces heat the quantity of which is also invariable. Thus, a pound of carbon, when carbon dioxide is formed, will always unite with $2\frac{2}{3}$ pounds of oxygen, and give off 14,600 B. T. U. As an intermediate step the carbon might unite with $1\frac{1}{3}$ times its weight of oxygen, and produce 4,450 B. T. U., but in its further conversion to CO_2 it would unite with an additional $1\frac{1}{3}$ times its weight of oxygen and evolve the other 10,150 B. T. U., since the heat developed in any chemical combination depends upon the initial and final states, and not upon any intermediate change.

Calorific Value of Fuel—The amount of heat liberated per pound of fuel undergoing *perfect combustion* is called the calorific value of the fuel. The methods of determining the calorific value will be treated in chapter on Determination of Heating Value of Fuels.

*See foot-note, page 112. †*Marine Boiler Management and Construction*, page 93.

Table 26 gives calorific values, air required, etc., for the elementary combustibles and several compounds.

Hydrogen Available for Combustion—During complete combustion the carbon will be converted into carbon dioxide, (CO_2); the hydrogen into water vapor, (H_2O); and the sulphur into gas of the composition SO_2. Not all the hydrogen shown by a fuel analysis is, however, available for heat production, since the oxygen shown by the analysis was united with part of the hydrogen in form of water, hence was already in combination before combustion was effected. Since water is H_2O and the atomic weights of H and O are respectively 1 and 16, the weight of combined hydrogen will be one-eighth of the weight of the oxygen, hence the hydrogen *available for combustion* will be $H-\frac{1}{8}O$.

Carbon	74.79%
Hydrogen	4.98
Oxygen	6.42
Nitrogen	1.20
Sulphur	3.24
Water	1.55
Ash	7.82
	100.00%

Substituting in the formula

$$\text{B. T. U. per pound} = 14600 \times 0.7479 + 62000 \left\{ 0.0498 - \frac{0.0642}{8} \right\} + 4000 \times 0.0324 = 13{,}650, \text{ very nearly.}$$

A calorimeter test showed 13480 B. T. U. for this coal, which illustrates the degree of accuracy to be expected of the formula. A more refined computation would involve a correction for heat expended in evaporating the water in the coal and superheating the resultant vapor to the breeching temperature.

TABLE 26

COMBUSTION DATA FOR CARBON, HYDROGEN, ETC.

1 Oxidizable Substance, or Fuel.	2 Chemical Symbol	3 Atomic or Combining Weight	4 Chemical Reaction.	5 Product of Combustion.	6 Oxygen per lb. of Col. 1 lbs.	7 Nitrogen per lb. of Col. 1 = 3.32 × O lbs.	8 Air per lb. of Col. 1 = 4.32 × O lbs.	9 Gaseous Product per lb. of Col. 1 = Col. 1 × Col. 8. lbs.	10 Heat Value per lb. of Col. 1 B. T. U.
Carbon	C	12	$C+2O=CO_2$	Carbon Dioxide	$2\frac{2}{3}$	8.85	11.52	12.52	14,600
Carbon	C	12	$C+O=CO$	Carbon Monoxide	$1\frac{1}{3}$	4.43	5.76	6.76	4,450
Carbon Monoxide	CO	28	$CO+O=CO_2$	Carbon Dioxide	4/7	1.90	2.47	3.47	10,150*
Hydrogen	H	1	$2H+O=H_2O$	Water	8	26.56	34.56	35.56	62,000
Methane	CH_4	16	$CH_4+4O=CO_2+2H_2O$	Carbon Dioxide and Water	4	13.28	17.28	18.38	23,550
Sulphur	S	32	$S+2O=SO_2$	Sulphur Dioxide	1	3.33	4.32	5.32	4,050

Dulong's Formula—The heating value of the various elements being known, the following formula, due with slight modifications to Dulong, enables the heat of combustion of a fuel to be computed.

Heating value of fuel per pound=

$$14{,}600C + 62{,}000 \left\{ H - \frac{O}{8} \right\} + 4{,}000S \qquad [24]$$

$C, H, O,$ and S are the proportionate parts, by weight, of carbon, hydrogen, oxygen and sulphur, respectively. The formula does not apply when the fuel contains carbon monoxide, (CO), but can be made to apply by adding the term 10,150 C†, in which C is the proportionate weight of carbon which is converted into carbon monoxide.

Assume, for example, a coal whose composition is as follows:

Air Required—From Table 26 the air required can be readily calculated, thus:

$$0.7479\ C \times 2\tfrac{2}{3} \ldots = 1.9944 \text{ lbs. O needed}$$

$$\left\{ 0.0498 - \frac{0.0642}{8} \right\} H \times 8 = 0.3262 \text{ '' '' ''}$$

$$0.0324\ S \times 1 = \ldots = 0.0324 \text{ '' '' ''}$$

$$\text{Total} \ldots = 2.3530 \text{ lbs. O needed}$$

Since 1 pound of oxygen is contained in 4.32 lbs. of air, the total air needed will be $2.353 \times 4.32 = 10.165$ lbs. The weight of actual combustible is

$$.7479 + .040775 + .0324 = .82 \text{ pounds,}$$

hence the air required per pound of combustible is $10.165 \div .82 = 12.4$ lbs.

*Per pound of carbon in the monoxide; the heat value of a pound of carbon monoxide is 4,350 B. T. U.

†Or by adding the term 4,350 CO if the weight of CO be known.

The air may be also found by following approximate formula:

Weight of air per lb. of fuel=

$$12C+36\left\{H-\frac{O}{8}\right\}+S \qquad [25]$$

in which the letters have same significance as in Dulong's formula above given.

Table 27 gives the air supply for various fuels, calculated as above explained.

TABLE 27

CALCULATED NET QUANTITY OF AIR REQUIRED FOR COMBUSTION OF VARIOUS FUELS

FUEL.	WEIGHT OF GIVEN CONSTITUENT IN 1 LB. OF FUEL. CARBON. %	HYDROGEN %	OXYGEN. %	POUNDS OF AIR REQUIRED PER LB. OF FUEL.
Wood Charcoal	93.	...	....	11.16
Peat Charcoal	80.	...	....	9.6
Coke	94.	...	....	11.28
Anthracite Coal	91.5	3.5	2.6	12.13
Dry Bituminous Coal	87.	5.0	4.0	12.06
Lignite	70.	5.0	20.0	9.30
Dry Peat	58.	6.0	31.0	7.68
Dry Wood	50.	...	....	6.00
Mineral Oil	85.	...	....	15.65

The above values are useful for comparison with the air actually used in any given case. To produce perfect combustion with the calculated quantity of air would require that each particle of oxygen be brought into intimate contact with the fuel. This cannot be done in practise, because of the mixture of the oxygen with nitrogen in the air, the irregular thickness of the fire, and varying resistance to passage through the fuel in different places owing to ash, clinker, etc. Where such difficulties are absent, as when burning gas or oil fuel, the air supplied may be materially less than that required for coal. Experiment shows that under either natural or forced draft coal requires about 50% more than the net calculated amount of air, or about 18 pounds per pound of coal. If less is supplied the carbon burns to monoxide instead of dioxide, thus fails to develop its full heat value. An excess of air is also a source of waste, as it dilutes the products of combustion, and reduces the temperature by absorbing heat which is conveyed to the breeching. Table No. 28 by Coxe* indicates the magnitude of the heat losses due to excess of air supply. By *minimum air supply* is meant the net calculated quantity,

Temperature of the Fire—If the heat due to combustion of the fuel and the weight and specific heat of the products of combustion be all known, the temperature of the furnace (neglecting heat lost by radiation and conduction) can be calculated. Evidently,

Heat of combustion in B. T. U=Weight of products of combustion × their specific heat × elevation of temperature of products in degrees. [26]

TABLE 28

SHOWING HEAT LOSSES WHEN BURNING 100 POUNDS OF ANTHRACITE WITH MINIMUM, AND TWICE THE MINIMUM AIR SUPPLY

(100 pounds of anthracite are assumed equal to 1,313,080 B. T. U. T=chimney temperature. Atmospheric temperature assumed at 60° F.)

	TOTAL HEAT LOST IN GASES. T=400°. B. T. U.	%	T=500°. B. T. U.	%	T=600°. B. T. U.	%	T=700°. B. T. U.	%	T=800°. B. T. U.	%
Minimum Air Supply	145,755	11.0	171,754	13	197,753	15	223,751	17.0	249,751	19.0
Twice Minimum Air Supply	230,097	17.5	279,007	21	329,176	25	378,715	28.8	428,254	32.6

To illustrate, assume that the same coal as in last problem is burned with the minimum supply of air. The sulphur and the small quantity of oxygen needed to burn it may be neglected. Then:

*See Thurston, *Manual of Steam Boilers*, p. 672.

Weight of CO_2=.7479+1.9944 =2.7423 lbs.
Weight of H_2O=0.04077+ 0.3262 . . . . = .3310 "
Nitrogen carried in by 10.165 lbs. of air=10.165×0.7685 . =7.8118 "
Total weight of products, including nitrogen . . . . 10.8861 "

combustion with the minimum quantity of air, such a furnace temperature cannot possibly be reached in practise. To illustrate the diminution of furnace temperature due to excess of air supply, the preceding case will be worked out on basis of twice the minimum air supply. This gives the following result:

TABLE 29

COOLING EFFECT OF VARIOUS PERCENTAGES OF EXCESS AIR

(Based on coal containing C=85%; H=2.5%; N=1%; Ash=7.75%, and B.T.U.=14,750 per pound. Temperature of external air=0°F.)

PER CENT. OF AIR ADDED TO THE MINIMUM QUANTITY.	IDEAL TEMPERATURE OF COMBUSTION. DEGREES.	LOSS OF TEMPERATURE DUE TO DILUTION. DEGREES.	TEMPERATURE OF COMBUSTION COMPARED WITH THAT DEVELOPED BY MINIMUM QUANTITY OF AIR.
0 (or Minimum Quantity)	5,132° F	......	.....
10% Added	4,710	422	91.8%
20	4,352	780	84.8
30	4,044	1,088	78.8
40	3,777	1,355	73.6
50	3,543	1,589	69.0
60	3,336	1,796	65.0
70	3,153	1,979	61.4
80	2,988	2,144	58.2
90	2,840	2,292	55.3
100	2,705	2,427	52.7
125	2,419	2,713	47.1
150	2,188	2,944	42.6
175	1,997	3,135	38.9
200	1,837	3,295	35.8

The mean specific heat of this mixture being unknown, the heat necessary to raise the mixture one degree will be now calculated.

2.7423 × 0.217 (specific heat of CO_2 . . . . =0.59508 B. T. U.
0.3310 × 0.4805 (specific heat superheated steam.*= 0.15905 "
7.8118 × 0.2438 (specific heat of N) . . . . = 1.90452 "
Hence total heat to raise products one deg. F. = 2.65865 B. T. U.

The calculated calorific value of this coal was 13650 B. T. U., hence the elevation of temperature will be $\frac{13650}{2.65865}$ = 5134° F. But because of the impossibility of getting proper

B. T. U. for minimum air supply as already determined . . . . = 2.65865
Additional B. T. U. for 10.165 lbs. of air = 10.165 x 0.2375 . . . = 2.41419
B. T. U. required to raise products of combustion (including nitrogen) one degree . . . . = 5.07284

Hence elevation of furnace temperature= $\frac{13650}{5.07284}$ = 2690° F.

In the preceding computation the specific heat of air has been assumed as equal to .2438, its value at 32°F., as is almost invariably done when computing furnace temperatures. It is known, however, that at high temperature the specific heat of air is

(*) Heat to raise the water to boiling point is here neglected. The specific heat of superheated steam is also greater than here used. See p. 93.

considerably greater, and while exact determinations have not yet been made, enough has been done to show that at the usual furnace temperatures the specific heat closely approximates to 0.3. Assuming this value, the preceding computation would give a furnace temperature of 2220° which is more nearly correct than the temperature of 2690° previously found.

750 H. P. OF STIRLING BOILERS, ROBINSON'S CENTRAL DEEP, LIMITED, SOUTH AFRICA

Table 29 further illustrates the cooling effect of excess air. In practical work the temperatures will be even less than given in the table, because of losses due to radiation, slicing fires and removal of ashes, and further fact that at high temperatures the specific heat of gases is probably greater than the values for lower temperatures, the only values at present available for use in making the computation.

The temperature is also lowered by insufficient air supply because the carbon will be burned only to carbon monoxide, which will develop less than one-third of the heat which is produced when the carbon is converted into carbon dioxide.

This subject is of greatest practical importance, owing to the large and usually unsuspected loss of fuel due to incompetent or careless firing. Large sums of money are often spent on devices intended to save a few per cent. of fuel, while through careful attention by a skilled fireman much greater savings could be effected without any expense whatever. The computations are also valuable as indicating why boiler efficiencies, when gas or oil fuel is used, are often ten to fifteen per cent. higher than when burning coal, the difference being due to decrease in the excess of air used, and prevention of heat losses due to hot ashes, slicing fires, opening of fire doors, and admission of cold air into the furnace when burning coal.

LEHIGH PORTLAND CEMENT COMPANY ALLENTOWN, PA., OPERATING 10,500 H. P. OF STIRLING BOILERS

Fuels for Steam Boilers

Fuels may be solid, liquid, or gaseous. Such representatives of each class as are used for firing steam boilers will be considered.

Coal is the fossilized remains of prehistoric vegetable growth. In its stages from vegetable to almost pure carbon in the form of graphite, it was successively changed into the forms listed in Table 30. With each stage the content of carbon increases.

TABLE 30

APPROXIMATE CHEMICAL CHANGES, WOOD FIBRE TO ANTHRACITE COAL

SUBSTANCE.	CARBON.	HYDROGEN.	OXYGEN.
	%	%	%
Wood Fibre	52.65	5.25	42.10
Peat	59.57	5.96	34.47
Lignite	66.04	5.27	28.69
Earthy Brown Coal	73.18	5.58	21.14
Bituminous Coal	75.06	5.84	19.10
Semi-bituminous Coal	89.29	5.05	6.66
Anthracite Coal	91.58	3.96	4.46

Table 31 gives the approximate percentages of carbon and volatile matter in the combustible portion of the general classes of coals.

TABLE 31

CLASSIFICATION OF COALS ACCORDING TO CONSTITUENTS IN THE COMBUSTIBLE*

	FIXED CARBON.	VOLATILE MATTER.
Anthracite	97 to 92.5%	3 to 7.5%
Semi-anthracite	92.5 to 87.5	7.5 to 12.5
Semi-bituminous	87.5 to 75	12.5 to 25
Bituminous, Eastern	75 to 60	25 to 40
Bituminous, Western	65 to 50	35 to 50
Lignite	under 50	over 50

The percentages of ash and moisture in coal vary greatly. The ash ranges from three to thirty per cent., and the moisture from 0.75 to 25 per cent. of the total weight of the coal, depending upon the locality where mined and the grade.

Anthracite, or Hard Coal, ignites slowly, but when in a state of incandescence its radiant heat is very great. The name Anthracite may be applied to all those dry or non-bituminous coals which, possessing from three to seven per cent. of a gaseous matter, do not swell when burned. Its flame is quite short and of a yellowish blue tinge and it can be burned with practically no smoke. True or dry anthracite is characterized by few joints and clefts, and their squareness; great relative hardness and density; high specific gravity, ranging from 1.4 to 1.8: and semi-metallic luster.

Anthracite is now classed and marketed according to sizes, the following division of mesh being adopted as standard at Wilkesbarre in 1891:

Egg Coal must pass through 2⅝" mesh and not through 2"
Stove " " " " 2" " " " " 1⅜"
Chestnut " " " 1⅜" " " " " ¾"
Pea Coal " " " ¾" " " " " ½"
Buckwheat No. 1 must pass through ½" mesh and not through ¼"
Buckwheat No. 2 or rice, must pass through ¼" mesh and not through ⅛".

Semi-anthracite coal, because of its content of seven to twelve per cent. of volatile combustible, kindles more readily and burns more rapidly than anthracite. It has less density, hardness, and metallic luster than anthracite and the usual specific gravity is about 1.40.

Semi-bituminous coal is softer, contains more volatile matter, kindles easier and burns more rapidly than anthracite. It gives an intense and free burning fire.

Bituminous coals range in color from pitch black to a dark brown. Their luster is resinous or vitreous in the most compact specimens, and silky in those showing traces of vegetable fibre. The specific gravity is usually about 1.3. The distinctive characteristic of the bituminous coals is the emission of yellow flame and smoke when burning.

Bituminous coals absorb moisture from the atmosphere. The surface moisture can be removed by ordinary drying, but a large portion of the water can be separated from the coal only by heating it to a temperature of about 250° F.

Bituminous coals are either caking or non-caking. The former when heated fuse together and swell in size; the latter burn freely, do not fuse and are commonly known as "free burning" coals. Caking coals are usually rich in volatile hydrocarbons and are valuable for gas manufacture.

*Kent's *Steam Boiler Economy*, page 42.

Cannel Coal is a variety of bituminous coal, rich in hydrogen and hydrocarbon, and is exceedingly valuable as a gas coal; it is bright flaming, burns without melting, and has a dull resinous luster. It is seldom used for steaming purposes, although it is sometimes mixed with Pocahontas coal when increased economy is desired at very high combustion rates. Cannel coal usually shows the following composition:

Fixed carbon	26 to 55%
Volatile Matter	42 to 64
Earthy Matter	2 to 14

The specific gravity is about 1.24.

Lignite is vegetable matter in the earlier stages of conversion into coal. Its specific gravity is low, 1.2 to 1.23, and when freshly mined it contains as high as fifty per cent. of moisture. Its appearance is not uniform, and varies from a light brown color of distinctly woody structure to specimens resembling hard coal. It is easily broken, will not stand much handling in transportation, rapidly absorbs moisture and if exposed to the weather it splits up into fine pieces like air slacked lime, which greatly increases the difficulty of burning it. It is non-caking and gives a bright but slightly smoky flame with moderate heat.

Its composition is extremely variable, even in the same deposit; the ash may run as low as one per cent. and as high as fifty-eight per cent. The high content of moisture and large amount of air necessary for burning lignite cause large stack losses, and in consequence it is a low grade fuel.

Composition of Coal—The uncombined carbon in coal is known as *fixed carbon*. There is also some carbon combined with hydrogen, and this, together with other gaseous substances driven off by the application of heat, constitutes the volatile portion of the fuel. The fixed carbon and the volatile matter constitute the combustible,* the other important ingredients entering into the composition of coal being *moisture*, and the refractory earths which form the *ash*.

A large percentage of ash is undesirable, because it not only reduces the calorific value of the fuel, but in the furnace clogs up the air passages and prevents the rapid combustion necessary to high efficiency. If the coal also contains an excessive quantity of sulphur, trouble will be experienced because sulphur is not only injurious to boiler steel, but unites with the ash to form a fusible slag or clinker which chokes up the grate bars and forms a solid mass, having imbedded in it large quantities of unconsumed carbon. Moisture in coal is more detrimental than ash in lowering furnace temperatures, because it is not only non-combustible, but it absorbs heat when it evaporates and is superheated to the temperature of the stack gases.

Coal Tables — The properties of the various classes of coals in the progression from lignite to anthracite are shown in Table 32. Data pertaining to the composition and

TABLE 32

APPROXIMATE HEATING VALUE OF GENERAL GRADES OF COAL PER POUND OF COMBUSTIBLE, B. T. U.

KIND OF COAL.	PER CENT. OF COMBUSTIBLE.		HEATING VALUE PER POUND OF COMBUSTIBLE.
	FIXED CARBON.	VOLATILE MATTER.	
Anthracite	97.0 to 92.5	3.0 to 7.5	14,600 to 14,800
Semi-anthracite	92.5 to 87.5	7.5 to 12.5	14,700 to 15,500
Semi-bituminous	87.5 to 75.	12.5 to 25.	15,500 to 16,000
Bituminous, Eastern	75. to 60.	25. to 40.	14,800 to 15,200
Bituminous, Western	65. to 50.	35. to 50.	13,500 to 14,800
Lignite	Under 50	Over 50	11,000 to 13,500

*The oxygen and nitrogen contained in the volatile matter are not really *combustible*, but through custom the term combustible is generally applied to that part of the coal which is dry and free from ash, which includes the oxygen and nitrogen.

calorific value of the principal coals in the United States are presented in Table 33. Preparation of this table has been difficult because of the dearth of reliable calorimeter tests on fuels from all the various localities. Published results are often unreliable because of failure to specify whether these results apply to dry coal, or coal in its natural state, and also because of doubt as to the degree of accuracy of the calorimeter used. In many cases it has been necessary to compute the calorific value, but the results thus obtained are probably as nearly accurate as most of the others given, because of the variation in quality of samples, and discordant results given by different calorimeters for samples taken from the same seam of coal. The tabular values are therefore to be regarded as approximations only, and in any important case a properly selected sample of the coal under consideration should be submitted to a competent chemist for determination of the calorific value.

Attention is called to the difference between the calorific value of coal per pound of combustible and per pound of fuel. If a coal contains ninety per cent. of combustible, and has a thermal value of 14,500 B. T. U. per pound of combustible, the heating value per pound of coal will be

$$14{,}500 \times 0.90 = 13{,}050 \text{ B. T. U.}$$

In Table 33 only the calorific values per pound of combustible are given, and the value per pound of coal containing any given per cent. of ash and moisture can be quickly computed as above shown.

Weathering of Coal produces results which vary with the kind of coal. Anthracite is but little affected, apart from the oxidation of the sulphur content, which is small. Since bituminous coals usually contain a higher percentage of sulphur than occurs in the anthracite, weathering will produce more rapid oxidation, which frequently generates sufficient heat to cause spontaneous combustion. When this occurs the unaffected coal should at once be removed, and the heated coal be spread so that there may be a free circulation through it. When it is necessary to pile coal containing much sulphur, or expose it to the weather, the risk of spontaneous combustion may be diminished by making the pile as shallow as possible, and by inserting into it, at intervals of six to seven feet, pieces of pipe which stand vertically, and are open top and bottom, so as to promote the free circulation of air through the mass.

Weathering destroys the coking property of coals; it also rapidly disintegrates some lignites and renders them difficult to burn on an ordinary grate.

Sufficiently complete experiments to determine the relation between weathering of coal and the decrease in its heat value have not yet been made. The experiments thus far available indicate that there is some loss; probably the loss will be greater as the per cent. of volatile matter in the coal is increased. Besides this, the augmented difficulty of burning weathered coals of low grade must be considered.

It is bad practise to pile coal on the bare ground. When shoveled up, such coal will invariably be mixed with more or less dirt, gravel, etc., all of which promotes the formation of clinkers and destruction of grates.

Coal Dust—The utilization of dust, slack, and small sizes of coal that would otherwise go to waste has been the subject of considerable investigation and experimentation, resulting in numerous processes for briquetting the material and burning it in the form of lump, as before described; another method which has come into favor in some localities is to pulverize the coal into a dust and use it as a fuel in this form. From data at present available it seems that advantages may be expected from such a fuel, since the utilization of the combustible portion is more complete than with solid fuel, the production of smoke is minimized, and the process admits of an adjustment of the air supply to a point very close to the theoretical quantity. This is due to the intimate admixture of the air and fuel, and to the possibility of maintaining a more nearly uniform furnace temperature. The principal objections to the use of coal dust as a fuel are the liability of the feed pipes and passages adjacent to the furnace to choke up, the difficulty of reducing the fuel to a uniform degree of fineness, liability of explosions in the furnace, and the gathering of dust on the boiler heating surface, thereby diminishing its capacity and efficiency.

TABLE 33

APPROXIMATE COMPOSITION AND CALORIFIC VALUE OF PRINCIPAL AMERICAN COALS

LOCALITY WHERE MINED.	PROXIMATE ANALYSES.					Approximate Calorific Value in B. T. U. per lb. of Combustible*	Authority.
	Volatile Matter. Per Cent.	Fixed Carbon. Per Cent.	Moisture. Per Cent.	Ash. Per Cent.	Sulphur. Per Cent.		
PENNSYLVANIA ANTHRACITES.							
East middle field, Wharton bed . . .	3.08	86.40	3.71	6.22	0.58	15,000	(a) (b)
" " " Mammoth bed. . .	3.08	86.38	4.12	5.92	0.49	15,000	" "
West middle field, Buck mountain bed .	3.95	82.66	3.04	9.88	0.46	15,070	" "
" " " Seven foot bed . .	3.98	80.87	3.41	11.23	0.51	15,080	" "
" " " Primrose bed . .	3.72	81.59	5.54	10.65	0.50	15,060	" "
" " " Mammoth bed . .	3.72	81.14	3.16	11.08	0.90	15,060	" "
Northern field, Mammoth bed . . .	4.38	83.27	3.42	8.20	0.73	15,100	" "
Southern field, Mammoth bed . . .	4.28	83.81	3.09	8.18	0.64	15,100	" "
" " Primrose bed . . .	4.13	87.98	3.01	4.38	0.50	15,070	" "
Lykens Valley buckwheat	6.21	76.94	....	....	....	15,300	" (c)
" " "	6.8	80.2	....	....	....	15,400	" "
" " "	5.0	81.0	....	....	....	15,300	" "
PENNSYLVANIA SEMI-ANTHRACITE.							
Loyalsock	8.10	83.34	1.30	6.23	1.03	15,400	(a) (b)
Bernice basin	3.56	82.52	0.96	3.27	0.24	15,050	" "
" "	8.56	89.39	1.97	9.34	1.04	15,475	" "
SEMI-BITUMINOUS.							
Bradford County . . . . . . Penna.	16.95	69.26	0.82	12.29	0.67	15,800	(a) (d)
Sullivan County . . . . "	13.03	72.74	3.24	10.38	0.61	15,700	" "
Tioga County . . . . "	20.50	67.79	1.65	8.85	1.26	15,750	" "
Lycoming County . . . . "	17.53	72.42	1.06	8.15	0.84	15,800	" "
Center County . . . . "	22.60	68.71	0.60	5.40	2.69	15,700	" "
Huntington County . . . . . . "	13.84	78.46	0.79	6.00	0.91	15,700	" "
Blair County . . . . "	27.27	60.69	1.06	8.66	2.31	15,550	" "
Cambria County, lower bed . . . "	21.21	68.94	0.74	7.51	1.98	15,750	" "
" " upper bed . . . "	17.18	73.42	1.14	6.58	1.41	15,800	" "
Clearfield County, upper bed . . . "	23.94	69.28	0.70	4.62	1.42	15,700	" "
" " lower bed . . . "	21.10	74.08	0.81	3.36	0.42	15,800	" "
Somerset County . . . . "	19.77	67.78	1.15	9.67	1.61	15,800	" "
Broad Top . . . . "	17.38	76.14	0.78	4.81	0.88	15,800	" "
Cumberland . . . . Md.	19.13	72.70	0.95	6.40	0.78	15,820	" (e)
" . . . . "	15.47	73.51	1.23	9.09	0.70	15,820	" "
" . . . . "	15.52	74.28	0.89	9.29	0.71	15,800	" "
Pittsburg seam, George's Creek Valley "	19.19	74.91	0.59	5.31	0.63	15,840	" (f)
Bakerstown, " " " "	17.17	72.93	0.60	8.76	0.59	15,840	" "
Upper Freeport, " " " "	17.07	77.04	1.02	4.87	0.83	15,800	" "
Lower Kittanning, " " " "	16.92	76.58	0.74	5.86	0.68	15,840	" "
Pocahontas run of mine . . . . Va.	18.30	73.65	0.80	7.25	0.57	15,780	(g)
" " " " . . . . . "	18.62	75.12	0.63	5.63	0.57	15,720	"
" " " " . . . . . "	18.60	75.75	0.85	4.80	0.62	15,800	"

*Or coal dry and free from ash.

TABLE 33—Continued

LOCALITY WHERE MINED.		PROXIMATE ANALYSES. Volatile. Matter. Per Cent.	Fixed Carbon. Per Cent.	Moisture. Per Cent.	Ash. Per Cent.	Sulphur. Per Cent.	Approximate Calorific Value in B. T. U. per lb. of Combustible.	Authority.
New River district	W. Va.							
Quinnamont lump		18.65	79.26	0.76	1.11	0.23	15,820	(a) (h)
" slack		17.57	79.40	0.83	1.92	0.28	15,830	" "
Fire Creek		22.34	75.02	0.61	1.47	0.56	15,800	" "
Longdale		21.38	72.32	1.03	5.27	0.27	15,800	" "
Nuttalburg		25.35	70.67	1.35	2.10	0.57	15,720	" "
Hawk's Nest		21.83	75.37	1.93	1.87	0.26	15,800	" "
Ansted		32.61	63.10	1.40	2.15	0.74	15,350	" "
BITUMINOUS.								
Jefferson County	Penna.	32.53	60.99	1.21	3.76	1.00	15,300	(i)
Indiana County	"	29.26	58.74	0.98	9.46	1.73	15,400	"
Westmoreland County	"	32.27	59.23	1.14	5.97	1.50	15,200	"
Fayette County	"	29.75	60.47	0.95	7.04	1.79	15,400	"
Potter County	"	32.28	55.32	1.72	9.67	1.01	15,100	"
McKean County	"	34.49	46.25	2.25	14.02	2.97	14,600	"
Clarion County	"	38.60	54.15	1.97	4.10	1.19	14,700	"
Armstrong County	"	42.55	49.69	1.18	4.58	2.00	14,000	"
Butler County	"	39.88	48.97	1.91	7.22	1.97	14,200	"
Lawrence County	"	40.45	52.51	2.11	3.25	1.37	14,500	"
Beaver County	"	39.04	50.20	1.96	6.96	2.00	14,500	"
Washington County	"	37.11	50.99	1.16	8.72	2.06	14,700	"
Greene County	"	35.74	51.75	1.14	9.10	1.79	14,800	"
Youghiogheny River	"	36.49	59.05	1.03	2.61	1.81	15,100	"
Connellsville	"	30.10	59.61	1.26	8.23	0.78	15,400	"
Upper Freeport seam	Pa. and O.	37.35	51.63	1.93	9.10	2.89	14,750	(g)
Jackson County	Ohio.	35.79	52.78	8.17	3.25	1.13	14,140	"
Middle Kittanning, Hocking Valley	"	32.85	48.74	6.51	8.93	1.58	14,080	"
Mahoning Coal, Salinville	"	35.00	50.95	3.15	10.90	1.86	14,730	"
Massillon	"	31.83	64.25	2.47	1.45	0.56	15.075	(a)
Brier Hill	"	34.60	56.30	4.80	4.30	....	14,300	(k)
Big Stone Gap splint	Virginia.	33.90	59.25	1.80	5.05	0.71	15,100	(a) (c)
Carbon Hill	"	18.60	71.00	0.40	10.00	....	15,800	" "
Coal River	"	35.89	58.89	3.35	1.25	0.62	14,950	" "
Coal River splint	"	33.33	55.25	1.78	9.02	0.62	15,000	" "
Cedar Grove, Kanawha Co.	"	34.08	60.67	2.10	2.50	0.65	15,150	" "
Richmond coking coal	"	30.36	58.30	1.62	10.58	....	15,350	" "
South of James River	"	32.24	58.89	1.48	7.72	1.45	15,200	" "
Thacker	W. Va.	35.54	56.24	1.38	6.84	1.39	15,140	(g)
Coal Creek, Anderson County	Tenn.	34.86	58.41	1.29	5.44	0.20	15,050	(a) (c)
Etna, Marion County	"	23.72	63.94	0.94	11.40	1.19	15,700	" "
Franklin County	"	25.41	62.00	1.77	10.82	0.64	15,750	" "
Harriman	"	32.32	62.31	....	5.37	0.84	15,350	" "
Melville, Hamilton Co.	"	26.50	67.08	2.74	3.68	0.98	15,650	" "
Morgan County	"	34.55	61.66	1.67	2.14	0.88	15,150	" "

TABLE 33—Continued

LOCALITY WHERE MINED.		PROXIMATE ANALYSES.					Approximate Calorific Value in B. T. U. per lb. of Combustible.	Authority.
		Volatile Matter. Per Cent.	Fixed Carbon. Per Cent.	Moisture. Per Cent.	Ash. Per Cent.	Sulphur. Per Cent.		
Newcombe, Campbell Co.	Tenn.	33.77	60.64	....	3.59	1.20	15,200	(a) (c)
Rhea County	"	29.13	61.68	0.82	7.07	1.30	15,450	" "
Rockwood, Roane Co.	"	26.62	60.11	1.75	11.52	1.49	15,050	" "
Scott County	"	34.53	61.66	1.67	2.14	0.88	15,200	" "
Tracy City, Suanee Co.	"	29.30	61.00	1.60	7.80	....	15,450	" "
Boyd County	Kentucky.	33.77	54.51	3.27	8.91	1.56	14,950	(a) (l)
Carter County	"	34.60	55.25	4.10	4.77	1.41	14,850	" "
Coalton County	"	32.04	55.59	5.19	6.71	1.68	15,100	" "
Floyd County	"	36.70	51.70	1.30	10.30	1.36	14,400	" "
Greenup County	"	35.00	52.34	3.56	9.02	2.59	14,600	" "
Johnson County	"	38.04	56.30	2.66	3.00	1.29	14,600	" "
Lawrence County	"	35.70	53.28	4.60	6.42	1.08	14,600	" "
Martin County	"	32.60	62.68	1.46	3.26	....	15,300	" "
Pike County	"	26.80	67.60	1.80	3.80	0.97	15,600	" "
Bibb Co., Blockton, upper vein	Ala	34.01	59.51	2.28	3.25	0.45	15,150	(c)
Cahaba, Shelby County	"	33.28	63.04	1.66	2.02	0.53	15,400	(m)
Conglomerate	"	30.86	64.54	2.13	2.47	1.48	15,450	"
Helena	"	29.44	66.81	1.21	2.54	0.53	15,050	"
Pratt Co.'s Upper Jefferson Co.	"	32.29	59.50	1.47	6.73	1.22	15,250	"
Pratt Co.'s, Lower Jefferson Co.	"	30.68	63.69	1.53	4.10	0.61	15,450	"
Brazil	Indiana.	34.49	50.30	8.98	6.28	1.39	14,550	(a)
Block Coal	"	38.17	52.27	4.66	5.89	....	13,880	(n)
Big Muddy	Illinois.	31.9	53.0	7.7	7.4	....	14,550	(o)
Carterville mine run	"	34.11	52.17	4.87	8.85	0.85	14,150	(p)
" washed, No. 1	"	33.99	54.21	4.66	7.14	0.74	13,930	"
" " No. 2.	"	35.12	55.01	4.31	5.56	0.86	14,350	"
" " No. 4.	"	33.26	55.29	4.86	6.59	1.15	13,940	"
Collinsville, Madison Co.	"	45.89	31.57	9.20	13.34	5.34	13,080	"
Danville, Vermilion Co.	"	43.70	45.37	4.78	6.15	....	14,050	"
" " " screenings	"	33.80	34.20	9.40	23.10	....	13,760	"
Duquoin, Perry Co., lump	"	34.61	50.85	9.14	5.40	....	14,110	"
" " " nut	"	38.91	46.00	7.43	7.66	....	13,098	"
" " " slack	"	35.95	41.60	6.05	16.40	5.14	14,370	"
Glen Carbon, Madison Co., lump.	"	39.13	40.66	7.85	12.36	4.87	14,466	"
Girard, Macoupin Co.	"	34.39	45.76	9.70	10.15	3.50	12,410	"
La Salle	"	39.40	43.95	8.22	8.43	....	14,600	"
Mt. Olive	"	38.33	40.22	9.63	11.82	6.78	14,090	(o)
Mt. Olive	"	33.1	44.1	8.1	14.7	....	13,700	"
Pana, Christian Co., nut	"	39.43	46.04	5.30	9.23	....	13,860	(p)
Pana, Christian Co., slack	"	35.45	39.35	8.55	16.65	4.77	13,100	"
St. John, paradise lump	"	37.00	51.10	9.63	2.27	....	13,590	"
Stanton, Macoupin Co., lump	"	36.00	48.00	Dried	16.00	....	13,700	"
Streator, average	"	37.63	45.93	8.30	8.14	....	13,730	"
" lump	"	39.40	48.20	Dried	12.40	....	14,400	"

TABLE 33—Continued

LOCALITY WHERE MINED.		PROXIMATE ANALYSES.					Approximate Calorific Value in B. T. U. per lb. of Combustible.	Authority.
		Volatile Matter. Per Cent.	Fixed Carbon. Per Cent.	Moisture. Per Cent.	Ash. Per Cent.	Sulphur. Per Cent.		
Streator nut	Illinois	35.60	54.50	Dried	9.90	....	14,200	(p)
" screenings	"	38.40	43.80	Dried	17.80	....	14,100	"
Trenton, Clinton Co.	"	30.40	52.00	13.30	4.30	0.90	12,850	"
Vulcan, St. Clair Co., nut	"	30.86	45.09	7.44	16.61	1.30	12,440	"
Cleveland, Lucas Co.	Iowa.	39.76	42.12	6.66	11.48	....	11,660	(q)
Cincinnati Co.	"	26.58	40.03	23.99	9.13	....	14,850	"
Hiteman, Tyrone Co.	"	37.63	44.69	4.92	12.76	.....	11,480	"
Steam coal, Beacon Co.	"	35.64	38.09	4.09	20.37	....	12,830	"
Walnut block, Centerville	"	37.77	46.64	5.52	10.07	....	12,460	"
Smoky Hollow, Avery Co.	"	39.02	51.33	6.29	4.35	....	11,370	"
Whitebreast Fuel Co., Pekay	"	46.06	46.89	Samples all oven dried. Moisture in original coal varied from 4.03% to 17.47%; average was 8.08%.	7.05	2.81	14,020	(y)
Eldon Coal Co., Laddsdale	"	42.72	47.78		9.50	4.96	14,520	"
Mine No. 2, Hocking	"	45.18	45.34		9.48	3.98	13,870	"
Des Moines Coal Co., Marquisville	"	45.62	50.29		4.09	2.74	12,560	"
Lunsden Coal Co., Bloomfield	"	39.06	53.46		7.48	2.38	13,920	"
Whitebreast Fuel Co., Hilton	"	40.61	48.21		11.18	3.26	13,950	"
Block Coal Co., Centerville	"	37.79	54.85		7.36	3.29	13,690	"
Consolidation Coal Co., No. 10, Buxton	"	37.09	50.83		12.08	2.27	13,690	"
Crowe Coal Co., Boone	"	41.46	50.33		8.21	4.16	13,860	"
Corey Coal Co., Lehigh	"	37.98	47.98		14.04	5.90	14,460	"
D. Lodwick, Mystic	"	39.07	54.91		6.02	3.15	13,690	"
Platt Coal Co., Van Meter	"	40.54	51.04		8.42	3.68	13,390	"
Jasper Co. Coal Co., Colfax	"	42.24	50.27		7.49	3.08	13,110	"
Pittsburg	Kansas.	28.60	60.32	3.26	7.28	....	11,030	(q)
Weir City, No. 1	"	33.54	58.41	2.08	5.97	....	12,350	"
Weir City, No. 5	"	33.77	57.17	2.70	6.36	....	12,850	"
Central C. & C. Co., No. 66, Macon Co.	Mo.	39.10	41.83	12.00	7.07	3.44	12,580	(a) (r)
Central C. & C. Co., No. 70, Macon Co.	"	36.26	43.16	10.20	10.38	4.47	13,150	" "
Elliott Coal Co.. Randolph Co.	"	36.32	42.77	11.15	9.76	3.55	13,120	" "
Far. Consolidated, No. 6, Lafayette, Co.	"	36.14	44.70	11.95	7.21	2.57	12,890	" "
Marceline Coal Co., Linn Co.	"	33.25	47.27	9.45	10.03	5.73	13,420	" "
Mendota Coal Co., No. 2, Putnam Co.	"	34.11	39.85	17.59	8.45	3.21	11,840	" "
Murline Coal Co., Ray Co.	"	37.85	41.66	13.07	7.42	1.92	12,660	" "
Richmond & Camden Coal Co., Ray Co.	"	37.93	42.99	9.83	9.25	3.11	12,620	" "
Weir Coal Co., No. 3, Barton Co.	"	34.40	53.98	3.62	8.00	4.02	15,000	" "
Western Coal Co., No. 8, Barton Co.	"	35.73	53.72	2.35	8.20	4.10	15,020	" "
Hezron lump	Colorado.	37.52	54.39	Dried	8.09	....	12,970	(s)
Walsen run of mine	"	36.02	51.12	Dried	12.86	....	12,130	"
Bridgeport Coal Co., Wise Co.	Texas.	31.93	41.12	12.21	14.74	1.73	14,470	(t)
Cannel Coal Co., Webb Co.	"	48.84	36.61	3.46	11.09	2.09	14,080	"
Cisco, Eastland Co.	"	34.86	36.37	13.44	15.33	2.54	13,470	"
Eagle Pass, Maverick Co.	"	33.08	40.09	9.40	17.43	1.28	15,230	"
Rio Grande Coal Co., Webb Co.	"	47.95	38.89	4.09	9.07	2.45	12,720	"
Strawn, Palo Pintó Co.	"	31.78	42.04	4.00	22.18	2.39	15,600	"
Texas & Pacific Coal Co., Erath Co.	"	33.20	43.15	5.83	17.82	1.51	15,000	"

TABLE 33—Continued

LOCALITY WHERE MINED.	PROXIMATE ANALYSES.					Approximate Calorific Value in B. T. U. per lb. of Combustible.	Authority.
	Volatile Matter. Per Cent.	Fixed Carbon. Per Cent.	Moisture. Per Cent.	Ash. Per Cent.	Sulphur. Per Cent.		
LIGNITES AND LIGNITIC COALS							
Calvert Bluff, Robertson Co. . . Texas.	51.00	10.00	29.86	9.14	0.91	13,000	(t)
Como Coal Co., Hopkins Co. . . "	45.88	3.41	33.87	16.84	0.68	13,100	"
Glenn-Belto mine, Bastrop Co. . "	36.88	21.22	35.40	6.50	0.94	13,530	"
Houston Co. Coal Co. . . . . "	33.16	19.93	36.16	10.75	0.40	14,140	"
Lytle, Medina Co. . . . . . "	40.31	18.50	34.29	6.90	1.20	12,800	"
North Texas Coal Co., Wood Co. . "	45.21	11.60	35.60	7.59	0.47	13,460	"
Rockdale, Milam Co. . . . . "	34.26	22.73	34.72	8.29	1.04	13,100	"
Timson Coal Co., Shelby Co. . . "	39.53	23.05	31.96	5.46	1.46	12,870	"
Cumberland . . . . . Wyoming.	44.27	46.18	3.65	5.90	0.61	14,100	(u)
Hanna . . . . "	48.43	36.37	6.38	8.82	0.99	13,440	(v)
Kemmerer . . . . "	36.16	51.78	5.80	6.26	0.60	13,630	(u)
Rock Springs . . . . . . "	40.83	48.30	7.19	3.68	0.38	13,900	"
Black Diamond . . . . Colorado.	43.05	39.01	14.67	3.27	0.77	10,507	(c)
Erie . . . . "	32.71	45.98	18.57	2.74	0.52	11,360	"
Canon City, vertical . . . . "	37.61	51.36	7.01	4.03	1.02	13,097	"
" " upper . . . . "	36.74	47.93	6.56	8.76	0.62	11,170	"
" " lower. . . . . "	37.21	49.54	7.66	5.59	0.82	11,644	"
Golden City, 5 foot seam . . "	36.20	42.08	18.35	3.37	0.43	8,154	"
" " 12 " " . . . "	41.23	38.46	17.64	2.67	0.30	9,947	"
Gunnison River . . . . . "	12.16	84.65	1.50	2.29	0.70	14,240	"
Marshall . . . . "	37.84	46.43	13.19	2.54	0.66	11,478	"
Mount Carbon . . . . "	36.91	37.82	20.38	4.87	0.40	10,624	"
South Park . . . . "	33.79	58.62	6.30	1.28	0.47	12,204	"
Castle Gate . . . . Utah.	41.8	51.3	....	6.9	....	14,180	(w)
Black Diamond . . . . Washington.	43.18	49.81	4.32	2.69	0.76	13,600	(x)
Carbon Hill No. 4 Vein . . "	37.02	49.12	1.02	12.84	....	14,600	"
Occidental, No. 6, Renton . "	37.40	52.55	2.02	8.03	0.68	14,020	"
Renton Cooperative Co. No. 2 Vein "	37.38	53.60	3.44	5.58	0.73	13,830	"
Roslyn mine, No. 4. . . . "	38.20	49.40	1.90	10.50	0.41	14,590	"
Roslyn mine, Cle-Elum . . "	37.86	48.30	6.34	7.59	0.49	14,730	"
Skaget Coal & Coke Co. . . "	26.67	64.51	0.53	8.20	0.68	15,530	"
Wilkeson, C. & C. Co., No. 1 mine "	28.11	61.53	0.63	9.73	2.09	15,590	"

REFERENCES: (a) B. T. U. computed by Table No. 49 and Fig. 27. (b) Analyses from Kent's *Mechanical Engineer's Pocket-Book*, p. 625. (c) Selected analyses. (d) Kent's *Steam Boiler Economy*, p. 59. (e) Kent's *Pocket-Book*, pp. 625–6. (f) Analyses by W. B. Clark. (g) Lord and Haas' researches. (h) *Steam Boiler Economy*, p. 65. (i) *Ibid.*, p. 61. (k) *Steam Boiler Economy*, p. 46. (l) *Ibid.*, p. 65. (m) *Ibid.*, p. 67. (n) D. P. Jones. (o) *Steam Boiler Economy*, p. 73. (p) *Twentieth Annual Coal Report*, Illinois, 1902. (q) C. R. Richards. (r) *Sixteenth Report of Coal Mine Inspector*, Missouri, 1902. (s) Colorado Fuel and Iron Co. (t) *Bulletin No. 15, University of Texas.* (u) Union Pacific Coal Co. (v) Slosson and Colburn. (w) Carpenter. (x) *Third Biennial Report of Bureau of Labor*, State of Washington, 1901–2. (y) *Notes on Steam Generation with Iowa Coal*, by G. W. Bissell, M. E. in Bulletin No. 9, Engineering Experiment Station, Iowa State College.

Patent or Pressed Fuels—Among these may be classed fuels composed of the dust of some suitable combustible, pressed and cemented together by a substance possessing adhesive and inflammable properties. Such fuels, known as briquettes, are extensively used in France, and consist of carbon or soft coal, too small for ordinary commercial use, mixed with pitch and coal tar. They do not find much favor in this country, as the cost and difficulty of manufacture render them no more economical than coal.

Coke is produced in three ways: (1) From gas coal, in gas retorts; (2) From gas or ordinary bituminous coal, in special ovens. (3) From petroleum, by carrying the distillation of the residuum to a red heat. The process of manufacture necessitates expelling the hydrocarbon gases, hence coke is a porous product consisting almost entirely of carbon. It is a smokeless fuel, it readily attracts and retains water from the atmosphere, and if not kept under shelter it may absorb twenty per cent. of its weight of moisture.

TABLE 34

ANALYSES OF AMERICAN COKES

(Kentucky Geological Survey)

WHERE MADE.	NO. OF TESTS.	FIXED CARBON.	ASH.	SULPHUR.
		%	%	%
Connellsville, Pa.	3	88.06	9.74	0.810
Chattanooga, Tenn.	4	80.51	16.34	1.505
Birmingham, Ala.	4	87.29	10.54	1.195
Pocahontas, Va.	3	92.53	5.74	0.597
New River, W. Va.	8	92.38	7.21	0.562
Big Stone Gap, Ky.	7	93.23	5.69	0.749

Coke from gas works is usually softer and more porous than other kinds. It ignites more readily and burns with less draft than is required for the combustion of coke produced in ovens. It does not produce so intense a heat, hence it is not used extensively in factories where a smokeless fuel with great heat is needed. Oven coke is a dead gray black in color, porous and brittle, with a slightly metallic luster. It is used principally in blast furnaces, cupolas, smelting and other furnaces requiring a blast. Petroleum coke occurs in large irregular lumps, is a compromise in hardness between oven and gas-retort coke, blacker in color than either of the other classes and is used principally for making electric carbons.

Peat contains a large amount of water, averaging seventy-five to eighty per cent., and occasionally reaching ninety per cent. It is unsuitable for fuel until dried, and its composition then varies little from that of wood. The proportion in which the various primary constituents exist in dried peat is about as follows:

Carbon	58 to 60%
Hydrogen	6
Oxygen	30 to 31
Nitrogen	1.25 to 1.5
Ash	2.75 to 5

Some peats have been known to show eleven per cent. of ash. In computing the heat of combustion, it must be borne in mind that peat as usually dried in the air contains from twenty-five to thirty per cent. of water. While large deposits of peat are found in this country it has not thus far been found profitable to utilize them in competition with coal.

Wood is vegetable fiber which has not undergone geological changes, but usually the term is used to designate the compact substances familiarly known as tree trunks and limbs. When newly felled, wood contains moisture ranging from thirty to fifty per cent. by weight, and midway between these figures is considered a good average. After a year's ordinary drying in the atmosphere, the moisture is reduced to about eighteen to twenty-five per cent.

Wood is usually classified as hard and soft. Hard woods include oak, maple, hickory, birch, walnut and beech. Soft woods consist of pine, fir, spruce, elm, chestnut, poplar and willow. Hard woods give less heat per *pound* than soft woods, contrary to general opinion. Gottlieb's experiments proved that a pound of white pine has a heat value 8.25 per cent. more than that of white oak. Weber's experiments with fir (soft) and oak (hard) gave the following results:

	FIR.	OAK.
Carbon	51.08%	50.43%
Hydrogen	6.12	5.88
Oxygen and Nitrogen	42.90	43.69
Calorific Value, B. T. U.	8,790	7,440

From Table 36 it appears that about $1\frac{3}{4}$ lbs. of dry wood have the same calorific value as one pound of bituminous coal; also that

the heating value of the same weight of various woods does not vary over ten per cent. The table is based on *dry* wood, but woods in ordinary air-dried condition contain about twenty to twenty-five per cent. of moisture, hence the available heat producing content will be twenty to twenty-five per cent.

TABLE 35

RELATIVE CALORIFIC VALUE OF WOODS

WOOD.	SPECIFIC GRAVITY.	LBS. IN ONE CORD.	LBS. COAL EQUIVALENT TO ONE CORD OF WOOD.*
Hickory, shell bark	1.000	4469	1910
Oak, chestnut .	0.885	3955	1690
Oak, white . .	0.885	3821	1670
Ash, white . .	0.772	3450	1440
Dogwood . . .	0.815	3643	1560
Oak, black . .	0.728	3254	1390
" red . . .	0.728	3254	1390
Beech, white . .	0.724	3236	1380
Maple, hard (sugar)	0.644	2878	1230
Maple, soft . .	0.597	2668	1140
Cedar, red . .	0.565	2525	1080
Magnolia . . .	0.605	2704	1160
Pine, yellow . .	0.551	2463	1060
Sycamore . . .	0.535	2391	1020
Butternut . . .	0.567	2534	1090
Pine, New Jersey	0.478	2137	916
" pitch . .	0.426	1904	812
" white . .	0.418	1868	800
Poplar, Lombardy	0.397	1774	761
Chestnut . . .	0.552	2333	1000
Poplar, yellow .	0.563	2516	1080

less than in the table, and of the heat produced a part is absorbed in evaporating the water in the wood and superheating the steam thus formed. The heat so absorbed may be computed by formula, page 133, and the net calorific value of the wood may thus be determined if the per cent. of water is known. As a general average one per cent. of water will make a reduction of one and one-half per cent. in the heating value of wood. Since a pound of average bituminous coal is equal in evaporative power to about one and three-fourths pounds of dry wood, or about two and one-third pounds of wood containing twenty-five per cent. of moisture, the value of a cord of wood expressed in pounds of coal may with sufficient accuracy for practical purposes be taken from Table 35.

Spent Tan, which consists of the fibrous portion of the bark, is thirty per cent. by weight of the original oak bark. The calorific value of dry tan, containing fifteen per cent. of ash, is 6,100 B. T. U. per pound, while tan in the average state of dryness contains thirty per cent. of water and has a heating value of 4,284 B. T. U. The conditions for burning tan and all other similar wet fuels require that they be surrounded with heated surfaces and burning fuel, in order that they may be dried rapidly, and thorough combustion be secured.

Straw is one of the many inferior grades of fuel which are sometimes used when other fuels could be obtained only with difficulty and at greater cost. Table 37, on following page, gives the relative composition of wheat and barley air-dried straw.

Such straws have a calorific value of 5411 B. T. U. per pound according to Dulong's formula, [24] and weigh when pressed six to eight pounds per cubic foot. Experiments

TABLE 36

COMPOSITION AND CALORIFIC VALUES OF VARIOUS DRY WOODS (*Gottlieb*)

KIND OF WOOD	CARBON %	HYDROGEN %	NITROGEN %	OXYGEN %	ASH %	CALORIFIC VALUE B.T.U. PER LB.
Oak	50.16	6.02	0.09	43.36	0.37	8,316
Ash.	49.18	6.27	0.07	43.91	0.57	8,480
Elm	48.99	6.20	0.06	44.25	0.50	8,510
Beech	49.06	6.11	0.09	44.17	0.57	8,591
Birch	48.88	6.06	0.10	44.67	0.29	8,586
Fir	50.36	5.92	0.05	43.39	0.28	9,063
Pine	50.31	6.20	0.04	43.08	0.37	9,153
Poplar‡ . . .	49.37	6.21	0.96	41.60	1.86	7,834†
Willow‡ . . .	49.96	5.96	0.96	39.56	3.37	7,926†

*On basis 1 lb. coal = 2⅓ lbs. wood. ‡Values according to Chevandier. †Values by Formula No. 24.

in Russia show that winter wheat straw, dried at 230° F. gave a heating value of 6,290 B. T. U. when dry and 5,448 B. T. U. when containing ten per cent. of moisture.

Other straws gave a calorific value varying from 6,750 B. T. U. per pound for flax to 5,590 for buckwheat.

TABLE 37

COMPOSITION OF WHEAT AND BARLEY STRAWS

	WHEAT STRAW	BARLEY STRAW	MEAN VALUE
Carbon	35.86%	36.27%	36.00
Hydrogen	5.01	5.07	5.00
Oxygen	37.68	38.26	38.00
Nitrogen	0.45	0.40	0.50
Ash	5.00	4.50	4.75
Water	16.00	15.50	15.75

Bagasse, or Megass, is the name given to refuse sugar cane after the juice has been extracted. It is used largely on sugar plantations as a fuel for generating the steam required in operating the mills; upon the efficient use of bagasse as a fuel depends to a great extent the success of sugar raising as a financial proposition, particularly where the prices for sugar are low and the cost of coal delivered is high.

The heating value of bagasse depends mostly upon the fibrous matter of the cane. This varies in different countries but in general is greater as the age of the cane is increased. In Cuba, Hawaii, and the West Indies, the cane is left standing from twelve months to two years and the fibrous matter will run from eleven to twenty per cent. In Louisiana, where the frosts require the cane to be harvested after a life not exceeding six and one-half months, the weight of the fiber will not be more than nine or ten per cent. The tropical canes therefore have a much greater fuel value, and even with inefficient machinery the mills can be operated solely by the bagasse produced. It is very seldom that such results can be obtained where the fiber is less than twelve per cent. of the weight of the cane; consequently in the sugar mills of the United States supplementary boilers fired with coal are required. The economy of the mill is estimated by the number of pounds of coal or wood required per ton of sugar produced.

The average composition of Louisiana cane when ready to be ground is ten per cent. fiber and ninety per cent. juice. The juice consists of

85 % Moisture
10 % Sugar
5 % Molasses,

which makes the composition of sugar cane

76.5 % Moisture
9.0 % Sugar
4.5 % Molasses
10.0 % Fiber

In passing through the mills the content of juice extracted varies from 75% to 82% of the weight of cane, the average extraction being about 78%. Assuming 100 lbs. of cane, 78 lbs. of juice will be extracted, leaving 22 lbs. of bagasse, consisting of 10 lbs. of fiber and 12 lbs. of juice; hence taking into account the composition of the juice as above given, the 22 lbs. of bagasse consist of 10.2 lbs. moisture + 1.2 lbs. sugar + 0.6 lbs. molasses + 10.0 lbs. fiber.

Similarly, the composition of bagasse for other degrees of extraction may be computed, then reduced to percentages by weight of the bagasse itself, per following table:

	EXTRACTION		
	75%	78%	80%
Moisture	51	46.37	42.5
Sugar	6	5.45	5.0
Molasses	3	2.73	2.5
Fiber	40	45.45	50.0

Numerous experiments have shown the calorific value of the *fiber* contained in cane to be about 8,325 B. T. U. per pound of fiber; hence to obtain the calorific value of diffusion bagasse per short ton of cane, use the formula

$$\text{B. T. U. per short ton of cane} = \frac{2000 \times 8325 \times \%\text{ of Fiber}}{100} \qquad [27]$$

Obviously the formula gives only the heat generated by the bagasse, and not the *heat available* for steam generating. All moisture must be evaporated and the resulting vapor be superheated to the temperature of the stack gases. The heat so absorbed must be deducted from that calculated from the formula, to determine the available heat.

TABLE 38

FUEL VALUES OF ONE POUND OF DIFFUSION BAGASSE

Moisture in Bagasse. Per Cent.	Heat Developed Per Pound of Bagasse. B. T. U.	Heat Available Per Pound of Bagasse B. T. U.	Pounds of Bagasse Equivalent to 1 Pound of Coal Containing 14,000 B.T.U.	Estimated Temperature of Fire. Fahr.
0	8,325	8,325	1.68	2,465°
20	6,660	6,420	2.18	2,294
30	5,827	5,468	2.56	2,186
40	4,995	4,516	3.10	2,049
50	4,162	3,563	3.93	1,870
60	3,330	2,611	5.41	1,627
70	2,497	1,658	8.44	1,281
75	2,081	1,183	11.90	1,045

TABLE 39

VALUE OF ONE POUND OF MILL BAGASSE AT DIFFERENT EXTRACTIONS UPON CANE OF 10 PER CENT. FIBRE, AND JUICE CONTAINING 15 PER CENT. SOLID MATTER

PER CENT. EXTRACTION OF WEIGHT OF CANE. (1)	PER CENT. MOISTURE IN BAGASSE (2)	FIBER. PER CENT. IN BAGASSE (3)	FIBER. FUEL VALUE B. T. U. (4)	SUGAR. PER CENT. IN BAGASSE (5)	SUGAR. FUEL VALUE B. T. U. (6)	MOLASSES. PER CENT. IN BAGASSE (7)	MOLASSES. FUEL VALUE B. T. U. (8)
90	0.00	100.00	8,325	....	...	....	...
85	28.33	66.67	5,550	3.33	240	1.67	116
80	42.50	50.00	4,162	5.00	361	2.50	174
75	51.00	40.00	3,330	6.00	433	3.00	209
70	56.67	33.33	2,775	6.67	482	3.33	232
65	60.71	28.57	2,378	7.15	516	3.57	248
60	63.75	25.00	2,081	7.50	541	3.75	261
55	66.12	22.22	1,850	7.78	562	3.88	270
50	68.00	20.00	1,665	8.00	578	4.00	278
45	69.55	18.18	1,513	8.18	591	4.09	284
40	70.83	16.67	1,388	8.33	601	4.17	290
25	73.67	13.33	1,110	8.67	626	4.33	301
15	75.00	11.77	980	8.82	637	4.41	307
0	76.50	10.00	832	9.00	650	4.50	313

PER CENT. EXTRACTION OF WEIGHT OF CANE	TOTAL HEAT DEVELOPED B. T. U. SUM OF COLUMNS 4, 6 AND 8. (9)	HEAT REQUIRED TO EVAPORATE THE WATER PRESENT. B. T. U. (10)	HEAT AVAILABLE FOR STEAM GENERATION. B. T. U. (11)	POUNDS BAGASSE REQUIRED TO EQUAL 1 LB. COAL CONTAINING 14,000 B. T. U. (12)	COAL EQUIVALENT PER TON OF CANE. POUNDS. (13)	TEMPERATURE OF FIRE. FAHR. (14)
90	8,325	...	8,325	1.68	119	2,465°
85	5,900	339	5,561	2.52	119	2,236
80	4,697	509	4,188	3.34	120	2,023
75	3,072	611	3,361	4.17	120	1,862
70	3,489	679	2,810	4.98	120	1,732
65	3,142	727	2,415	5.80	121	1,612
60	2,883	764	2,119	6.61	121	1,513
55	2,682	792	1,890	7.40	121	1,427
50	2,521	815	1,706	8.21	122	1,350
45	2,388	833	1,555	9.00	122	1,284
40	2,279	849	1,430	9.79	123	1,222
25	2,037	883	1,154	12.13	124	1,077
15	1,924	899	1,025	13.66	124	1,002
0	1,795	916	870	15.93	126	906

Table 38 contains data relating to the heat developed, available heat, etc., per pound of diffusion bagasse of different contents of moisture, assuming the temperature of the air as 80° F. and that of the stack gases as 420°.

Reference thus far to the calorific value of bagasse has been confined to *diffusion* bagasse. In the diffusion process the cane is chopped into small pieces and subjected to a series of soaking processes, and the resulting bagasse contains only fiber and moisture. Mill-bagasse is the refuse left after the cane has been passed through the rolls. Not all of the juice is extracted, and the bagasse contains fiber, moisture and juice. The juice contains combustible matter, viz., sugar and molasses. The composition of the bagasse will vary according to the per cent. extraction, as above shown. According to Dr. Atwater the calorific value of sugar is 7,223 B. T. U. per lb., and that of molasses (dry matter only) is 6,956 B. T. U. per lb. Thus a pound of mill-bagasse has a calorific value of

$$\frac{8{,}325 \times \%\ \text{Fiber} + 7{,}223 \times \%\ \text{Sugar} + 6{,}956 \times \%\ \text{Molasses}}{100} \quad [28]$$

and its *available* heating value will be this amount, less the B. T. U.'s necessary to drive off the contained moisture.

Table 39 gives data pertaining to mill-bagasse of various extractions. Coal of 14000 B. T. U. per pound is taken as a standard, and the coal ratios are obtained by dividing 14000 by the *available heat* per pound of bagasse. Coal equivalents *per ton of cane* are obtained by dividing the number of pounds of bagasse, resulting from the several extractions, by the coal ratio. Thus with an extraction of 75% there are 500 lbs. of bagasse per (short) ton of cane, and 500 ÷ 4.17 (the coal ratio for 75% extraction) = 120 pounds of coal, or the coal equivalent. The coal equivalents are practically the same for all degrees of extraction; in other words, sugar cane has an almost constant heating value, irrespective of how much juice is extracted from it. As the extraction grows less there is a greater weight of bagasse per ton of cane, and while a great part of this weight is due to water and is therefore non-combustible, the amount of juice left over is also greater and this contains combustible matter which more than compensates for the additional water present. Thus the coal equivalent of cane per ton actually *increases* as the degree of extraction grows less.

A difficulty, however, is that large quantities of moisture render it difficult to burn the bagasse except in special furnaces. As the content of moisture increases the temperature of the fire decreases, and there is danger of a point being reached where the gases from the fuel will refuse to ignite. In Table 39, the highest attainable temperature resulting from the combustion of dry bagasse is 2,465° F., and of bagasse of 75% extraction only 1862° F. The calculations are made on a basis of a pound of fiber requiring 12⅔ lbs. of air; sugar, 9¾ lbs., and molasses, 9⅓ lbs.; which are twice the theoretical amounts for complete combustion. The methods of burning bagasse will be treated in chapter on Fuel Burning.

Corn is used as a fuel in some states when the crop is very abundant and the selling price is low.

The following are calorimeter tests of various samples of corn:

TABLE 40

CALORIFIC VALUE OF CORN (*Richards*)

MATERIAL.			HEATING VALUE IN B. T. U.		
			PER POUND OF MATERIAL.	PER POUND OF DRY MATERIAL.	PER POUND OF DRY COMBUSTIBLE.
Yellow	Dent	Corn and Cob	8,040	.....	.....
"	"	"	8,202	8,959	9,085
"	"	Cob . . .	7,214	7,841	7,958
White	"	Corn and Cob	7,841	.....	.....
"	"	Corn . . .	8,382	9,100	0,301
"	"	Cob . . .	7,571	8,174	8,285

Petroleum is practically the only oil which is sufficiently abundant and cheap to be used as a fuel under boilers. It possesses many advantages over solid fuels, and its use is on the increase.

Gasoline, Benzine, Kerosene, and other liquid oils distilled from petroleum are excellent fuels, but are too costly for use under boilers. The residuum after these have been distilled off is valuable as fuel.

There are three kinds of petroleum in use, namely those which on distillation yield; (1) Paraffin; (2) Asphalt; (3) Olefin. To the first group belong the oils of the Appalachian Range and middle West. They

are dark brown with greenish tinge. Upon distillation they yield such a variety of light oils that their value is too great to permit their general use as fuels.

To the second group belong the oils from California and Texas. These vary from reddish brown to jet black, and are used mostly for fuel.

The third group comprises oils from Russia, which are also used more extensively for fuel than for any other purpose.

contracts for purchase of oil should limit the content of water, else sufficient tankage should be provided to enable most of the water to be settled out of the oil before it is burned. A large content of water also causes trouble with the burners.

Gasoline Test—The content of water in fuel oil is often determined as follows: A burette or other tall vessel provided with glass stopper and graduated into 200 divisions is filled to the 100 mark with gasoline

TABLE 41

CALORIFIC VALUE OF CALIFORNIA OILS

KIND OF OIL.	Per Cent of Sulphur.	Per Cent of Silt.	Specific Gravity at 60° F.	Per Cent of Moisture.	CALORIFIC VALUE, B. T. U. PER LB.	
					Oil as Fired.	Oil Freed of Water.
Whittier	0.975	0.031	0.9417	1.06	18,428.4	18,626
"	0.735	0.010	0.9430	1.06	18,478.8	18,677
"	1.010	0.024	0.9410	.74	18,567.0	18,705
"	0.960	0.010	0.9407	.42	18,578.7	18,657
Whittier and Los Angeles mixed	0.980	0.054	0.9530	4.93	17,791.2	18,714
" " " " "	0.955	0.048	0.9529	4.62	17,887.5	18,754
" " " " "	0.845	0.032	0.9637	8.71	16,904.7	18,518
" " " " "	0.840	0.024	0.9629	8.82	16,956.0	18,596
" " " " "	.....	.....	......	4.54	17,862.5	18,607
" " " " "	.....	.....	......	4.565	17,839.0	18,692
" " " " "	.....	.....	......	4.25	17,974.5	18,772
" " " " "	.....	.....	......	3.63	18,039.0	18,719
Los Angeles	.....	.....	......	9.00	17,035.0	18,610
" "	.....	.....	......	9.87	17,122.0	18,997
" "	.....	.....	......	9.16	17,241.0	18,979
" "	.....	.....	......	8.47	16,980.0	18,551
" "	.....	.....	......	7.55	18,356.0	19,855
Kern Valley	.....	.....	......	2.66	19,419.0	19,950
Fullerton	.....	.....	......	2.07	19,686.0	20,102

In general, crude oils consist mostly of hydrogen and carbon, but contain small percentages of sulphur, nitrogen, arsenic, phosphorus, and silt. They also contain a content of water varying from less than one per cent. up to 50 per cent., depending upon the care that has been taken to seperate out the water which accompanies the oil when pumped from the well. Here as in all other fuels, the percentage of water affects the available heat of the oil, hence

It is then filled to the 200 mark with the oil to be tested, which has first been slightly warmed. The two are thoroughly shaken together; any shrinkage below the 200 mark is made up by adding more oil, and the whole is then allowed to stand in a warm place (sometimes on an engine cylinder) for 24 hours. The water and dirt settle to the bottom, and the number of divisions of each give their respective percentages, by volume, of the total.

Calorific Value of Petroleum—Accurate data on this subject are scarce. A pound of oil free of water is usually considered to have a calorific value of from 18,500 to 22,000 B. T. U. Assuming the ultimate analysis of an average sample as Carbon 84%, Hydrogen 14%, Oxygen 2%, and allowing for the combination of the oxygen with its equivalent of hydrogen to form water, the composition becomes Carbon, 84%, Hydrogen, 13.75%, Water, 2.25%; and the heat value per pound of petroleum, free from water, is

Carbon .	$.84 \times 14{,}600 = 12{,}264$	B. T. U.
Hydrogen	$.1375 \times 62{,}100 = 8{,}625$	B. T. U.
	Total, 20,889	B. T. U.

Prof. James E. Denton gives the following data as a result of his experiments with Beaumont (Texas) crude oil:

Carbon	84.60%	Sulphur	1.63%	Flash Point	142° F.
Hydrogen	10.90%	Specific Gravity	.920	Burning Point	181° F.
Oxygen	2.87%			Cold test	6° F.

The Stirling Company has made various tests on boilers fired with California oil, samples of which were subjected to calorimeter test, with results as shown in Table 41.

Table 43 gives composition and calorific value of some oils as compiled from various sources.

The nitrogen in petroleum varies from .008 to 1.10% while that of sulphur varies from .07 to 3%.

TABLE 43

COMPOSITION AND CALORIFIC VALUES OF VARIOUS OILS

KIND OF OIL.	Per Cent of Carbon.	Per Cent of Hydrogen.	Per Cent of Oxygen.	Specific Gravity.	B. T. U per pound.
Heavy Oil, West Virginia . .	83.5	13.3	3.2	0.873	.....
Light Oil, West Virginia . .	84.3	14.1	1.6	0.8412	21240
Heavy Oil, Pennsylvania . .	84.9	13.7	1.04	0.886	19224
Light Oil, Pennsylvania . .	82.0	14.8	3.2	0.816	.....
Oil from Beaumont, Texas . .	86.8	13.0	0.0	0.920	.....
Oil from California	84.0	12.7	1.2	0.920	.....
Canada Crude	84.3	13.4	2.3	.....	20410
Ohio Crude	80.2	17.1	2.7	.....	21600
Oil from Parma, Italy . . .	84.0	13.4	1.8	0.786	.....
Oil from Hanover, Germany .	80.4	12.7	6.9	0.892	.....
Oil from Galicia, Austria . .	82.2	12.1	5.7	0.870	18416
Light Oil from Baku, Russia .	86.3	13.6	0.1	0.884	22628
Heavy Oil from Baku, Russia .	86.6	12.3	1.1	0.938	19440
Refuse of Oil " " " . .	87.1	11.7	1.2	0.928	22628
Oil from Java	87.1	12.0	0.9	0.923	.....

The analysis and calorific value of the principal crude and residuum oils given by Mr. Orde are:

TABLE 42

ANALYSES OF OILS

	CARBON.	HYDROGEN.	OXYGEN, ETC.	B. T. U.
	%	%	%	Per lb.
Texas . . .	85.66	11.03	3.31	19,240
Borneo . . .	87.8	10.78	1.24	18,830
Caucasus . .	84.94	13.96	1.25	18,610
Burmah . .	86.4	12.1	1.5	18,865

The Flash Point is the temperature at which an oil gives off inflammable gases. The flash points of the various oils are not given in the tables; in fact, data upon this subject are scarce. In general the light oils have a low flash point, while the heavy grades have a much higher flash point. This matter is of the utmost importance in determining the availability of oil as a fuel. The flash points of oils whose specific gravities are below 0.85 are generally below 60° F., while those of oils whose specific gravity exceeds 0.85 are usually above

60° F. There are, however, many exceptions to this rule, notably a certain Roumanian oil, whose specific gravity is 0.899, and whose flash point is 240° F. The danger of explosion increases as the flash point is lowered, and the utmost care should be taken to lessen the possibility of the light vapors becoming ignited. When proper precautions are taken the use of oil is almost as safe as the use of coal.

Gravity of Oils—Fuel oils are often valued according to their gravity as indicated on the Beaume hydrometer, but the gravity is by no means an accurate measure of the relative calorific value.

Petroleum as Compared with Coal—Petroleum possesses the following advantages over coal:

(1) Much lower cost for handling, as the oil is fed by simple, mechanical means, and cost of stoking, removing ashes, etc., is eliminated.

(2) For equal heat value oil occupies less space than coal, and the storage space may be at considerable distance from the boilers without detriment.

(3) Higher boiler efficiencies and capacities are obtainable, because the combustion is more perfect; the excess air supply is greatly lessened; the furnace can be kept at constant temperature because fires do not require cleaning nor furnace doors to be opened for firing; smoke can be wholly eliminated, and the boiler heating surfaces do not quickly foul with soot.

(4) Intensity of the fire can be almost instantly regulated to conform to the demand for more or less steam.

(5) Oil does not, like coal, deteriorate with age when stored.

(6) Great reduction in number of men necessary around the plant, and freedom from dust, dirt and smoke, with their damage to adjoining property.

The disadvantages of oil are:

(1) It must have a high flash point to minimize danger of explosions.

(2) City or town ordinances may impose onerous conditions regarding location and isolation of oil tanks.

(3) The boiler repair bill will be high unless the boiler is specially adapted to use of oil. Whenever unatomized oil strikes the boiler surface, a burn, blister, or bag is almost sure to form, and in boilers of the tubular, or Scotch Marine type, such bagging may cause a serious explosion. Owing to the intense temperature in the fire-box, local overheating and burning of plates are also common in boilers which are either deficient in circulation or deposit scale on their hottest surfaces. Constriction of circulation will also inevitably cause "steam pockets" which rapidly burn out the tubes. For these reasons certain types of boilers which seem fairly well adapted to use of coal need excessive repairs when fired with oil.

Many tables have been published which purport to show the number of barrels of oil equivalent to a ton of coal, and *vice versa*. For example, assuming a barrel of oil to weigh 310 lbs., and a pound of oil to contain 20,000 B. T. U., the following table can be computed.

TABLE 44

COMPARISON OF OIL AND COAL

B. T. U. PER LB. OF COAL.	LBS. OF COAL EQUAL TO ONE BARREL OF OIL.	BARRELS OF OIL EQUAL TO ONE SHORT TON OF COAL.
10,000	620	3.23
11,000	564	3.55
12,000	517	3.87
13,000	477	4.19
14,000	443	4.52
15,000	413	4.84

This and all similar computations based upon the relative calorific power of petroleum and coal are of no practical value, for the reason that when burning oil, an efficiency ranging as high as 83% can be obtained with large boilers of good design, while with poorer grades of coal and smaller boilers the efficiency may fall to 65% or lower. The efficiency of either fuel will depend upon the size of the boilers, the adaptation of their grates and furnaces to the particular fuel used, the degree of intelligence of the men in charge, and other similar factors. Table 45, reproduced from *Power*, January 1905, takes into account different boiler efficiencies, but it assumes a *fixed* calorific value for oil, 18,500 B. T. U., hence this table like

others of similar nature, while very useful as a rough guide, cannot enable one to compute the saving possible by substituting one fuel for the other. The reason for this is as follows: The saving to be made may depend upon the labor which can be dispensed with, the available space for fuel storage, and facilities for conveying the oil by a pipe line, the

TABLE 45

EQUIVALENT HEAT VALUES OF COAL AND FUEL OIL, ALSO FACTORS FOR THE REDUCTION OF PLANT ECONOMY PER POUND OF COAL TO EQUIVALENT FIGURES PER GALLON AND BARREL OF FUEL OIL

Boiler Efficiency with Fuel Oil.	Net Efficiency = Boiler Efficiency less Oil used by Burners and Pumps.	Net Pounds of Steam Evaporated per Pound of Oil per Hour from and at 212° Fahr.	Net Pounds of Steam Evaporated per Barrel Oil per Hour from and at 212° Fahr.	POUNDS OF WATER EVAPORATED PER POUND OF COAL PER HOUR FROM AND AT 212° FAHR.															
				5	6	7	8	9	10	11	12	5	6	7	8	9	10	11	12
				POUNDS OF OIL EQUAL TO ONE POUND OF COAL								BARRELS OF OIL EQUAL TO ONE TON OF COAL							
70	66	12.64	4247	.3955	.4746	.5538	.6329	.7120	.7911	.8702	.9493	2.354	2.825	3.296	3.767	4.238	4.709	5.175	5.651
71	67	12.83	4311	.3897	.4676	.54 6	.6235	.7014	.7794	.8573	.9353	2.319	2.783	3.248	3.711	4.175	4.640	5.103	5.567
72	68	13.02	4375	.3840	.4608	.5376	.6144	.6912	.7680	.8448	.9216	2.285	2.742	3.200	3.657	4.114	4.572	5.028	5.485
73	69	13.21	4439	.3785	.4542	.5299	.6056	.6813	.7570	.8304	.9084	2.252	2.703	3.154	3.605	4.054	4.506	4.956	5.406
74	70	13.40	4502	.3731	.4477	.5224	.5970	.6716	.7462	.8208	.8955	2.221	2.665	3.108	3.554	3.998	4.442	4.887	5.331
75	71	13.60	4569	.3676	.4411	.5147	.5882	.6618	.7352	.8088	.8823	2.188	2.626	3.064	3.501	3.939	4.377	4.8 5	5.252
76	72	13.79	4633	.3625	.4350	.5076	.5801	.6526	.7251	.7976	.8702	2.158	2.580	3.022	3.453	3.885	4.317	4.748	5.180
77	73	13.98	4697	.3576	.4291	.5007	.5722	.6437	.7153	.7868	.8583	2.120	2.555	2.980	3.406	3.832	4.258	4.683	5.109
78	74	14.17	4761	.3528	.4234	.4940	.5645	.6351	.7057	.7762	.8468	2.100	2.520	2.941	3.360	3.780	4.200	4.621	5.041
79	75	14.36	4825	.3481	.4178	.4874	.5571	.6267	.6963	.7660	.8352	2.072	2.487	2.907	3.316	3.731	4.145	4.559	4.974
80	76	14.55	4889	.3436	.4123	.4811	.5498	.6185	.6872	.7560	.8247	2.045	2.454	2.863	3.272	3.682	4.090	4.499	4.909
				VALUES OF H								VALUES OF J							
70	66	12.64	4247	.0404	.0593	.0692	.0791	.0890	.0988	.1087	.1187	849.5	707.8	606.7	530.8	471.8	424.7	386.0	353.9
71	67	12.83	4311	.0487	.0584	.0682	.0779	.0877	.0974	.1071	.1169	862.2	716.5	615.8	538.8	478.8	431.0	391.9	359.2
72	68	13.02	4375	.0480	.0576	.0672	.0768	.0864	.0960	.1056	.1152	875.0	729.1	625.0	546.9	486.0	437.5	397.6	364.5
73	69	13.21	4439	.0473	.0568	.0662	.0757	.0853	.0946	.1038	.1136	887.8	739.8	634.1	554.9	492.0	443.9	403.4	369.9
74	70	13.40	4502	.0466	.0559	.0653	.0746	.0839	.0932	.1026	.1119	900.4	750.[illegible]	643.1	562.7	500.2	450.2	409.2	375.1
75	71	13.60	4569	.0459	.0551	.0643	.0735	.0827	.0919	.1011	.1103	913.8	761.5	652.7	571.1	507.6	456.9	415.3	380.7
76	72	13.79	4633	.0453	.0544	.0635	.0725	.0816	.0906	.0997	.1088	926.6	772.1	661.8	579.1	514.7	463.3	421.1	386.0
77	73	13.98	4607	.0447	.0536	.0626	.0715	.0805	.0894	.0983	.1073	939.4	782.8	671.0	587.1	521.8	469.7	426.4	391.4
78	74	14.17	4761	.0441	.0529	.0617	.0705	.0794	.0882	.0970	.1059	952.2	793.5	680.1	595.1	529.0	476.1	432.8	396.7
79	75	14.36	4825	.0435	.0522	.0609	.0696	.0783	.0870	.0957	.1044	965.0	804.1	689.3	603.1	536.1	482.5	438.6	402.0
80	76	14.55	4889	.0429	.0515	.0601	.0687	.0773	.0859	.1945	.1031	977.8	8[illegible]4.8	698.4	611.1	543.2	488.9	444.4	407.4

Net efficiency is determined by deducting from boiler efficiency 4 per cent., representing steam used for oil burners and oil pumps.

One ton of coal weighs 2,000 pounds. One barrel of oil weighs 336 pounds. One gallon of oil weighs 8 pounds. One pound of oil contains 18,500 B. T. U.

Equivalent gallons of oil per kilowatt hour = H × pounds of coal per kilowatt hour.

$$\text{Equivalent kilowatt hours per barrel of oil} = \frac{J}{\text{pounds of coal per kilowatt hour.}}$$

TABLE 46

ANALYSES AND CALORIFIC VALUES OF NATURAL GAS FROM DIFFERENT LOCALITIES

LOCATION OF WELL.	ANALYSIS BY VOLUME.								ANALYSIS BY WEIGHT.								Calculated Calorific Value in B. T. U. per 1,000 Cubic Feet of Gas.	Calculated Calorific Value in B. T. U. per lb. of Gas.
	H	CH_4	CO	CO_2	N	O	Heavy * Hydro-carbons.	H_2S	H	CH_4	CO	CO_2	N	O	Heavy Hydro-carbons.	H_2S		
1. Anderson, Indiana	1.86	93.07	0.73	0.26	3.02	0.42	0.47	0.15	0.22	91.00	1.18	0.70	5.14	0.82	0.82	0.32	1,016,900	22,030
2. Kokomo, "	1.42	94.16	0.55	0.29	2.80	0.30	0.30	0.18	0.17	91.90	0.93	0.75	4.78	0.58	0.51	0.37	1,021,850	22,040
3. Marion, "	1.20	93.16	0.60	0.30	3.42	0.55	0.15	0.20	0.15	90.50	1.01	0.91	5.80	1.06	0.26	0.41	1,008,760	21,680
4. Muncie, "	2.35	92.67	0.45	0.25	3.53	0.35	0.25	0.15	0.28	90.67	0.73	0.68	6.20	0.69	0.43	0.32	1,003,750	21,820
5. Bloomfield, New York	....	82.41	....	10.11	4.31	0.28	2.94	....	....	67.00	....	22.25	6.10	0.45	4.20	....	925,550	16,820
6. Olean, "	....	96.50	0.50	....	....	2.00	1.00	....	....	93.90	0.75	....	....	3.75	1.60	....	1,018,490	22,640
7. Findlay, Ohio	1.64	93.35	0.41	0.25	3.41	0.39	0.35	0.20	0.20	90.84	0.69	0.67	5.83	0.76	0.60	0.41	1,010,580	21,830
8. Fostoria, "	1.89	92.84	0.55	0.20	3.82	0.35	0.20	0.15	0.23	90.47	0.94	0.69	6.50	0.52	0.34	0.31	100,900	21,730
9. St. Mary's,"	1.94	93.85	0.44	0.23	2.98	0.35	0.20	0.21	0.24	91.73	0.76	0.62	5.20	0.68	0.34	0.43	1,019,720	21,980
10. Barnes Well, near St. Ive, Butler Co., Pa.	6.10	75.54	trace	0.34	....	....	18.12	....	0.70	69.15	trace	0.85	....	....	28.30	....	1,117,100	22,850
11. Cherry Tree, Indiana Co., Pa.	22.50	60.27	....	2.28	7.32	0.83	6.80	....	3.26	62.70	....	6.52	13.40	1.72	12.40	....	841,500	19,270
12. Grapeville, Westmoreland Co., Pa.	7.05	35.08	0.22	0.58	27.87	0.16	29.04	....	0.63	25.50	0.27	1.16	35.50	0.24	36.70	....	892,950	14,270
13. " " " "	24.56	14.93	trace	trace	18.69	1.22	40.60	....	2.47	12.00	trace	trace	26.10	2.00	57.30	....	924,500	16,460
14. Harvey Well, Butler Co., Pa.	13.50	80.00	trace	0.66	....	....	5.72	....	1.83	86.50	trace	1.95	....	....	10.80	....	997,500	23,800
15. Leechburg, Westmoreland Co., Pa.	4.79	89 65	0.26	0.35	....	....	4.39	....	0.60	90.29	0.44	0.97	....	....	7.70	....	1,046,380	23,380
16. Pittsburg, Pa.	9.64	57.85	1.00	....	23.41	2.10	6.00	....	0.86	41.50	1.25	....	45.94	3.00	7.45	....	747.520	12,000
17. " "	14.45	75.16	0.30	0.30	2.89	1.20	5.40	....	2.09	83.09	0.58	0.91	0.50	2.43	10.40	....	947,730	23,250
18. " "	20.02	72.18	1.00	0.80	....	1.10	4.30	....	2.83	81.50	1.97	2.60	....	2.60	8.50	....	916,620	22,840
19. " "	26.16	65.25	0.80	0.60	....	0.80	6.30	....	3.88	77.50	1.66	1.96	....	1.90	13.10	....	898,730	23,310
20. " "	20.03	60.70	0.58	....	....	0.78	8.90	....	4.40	73.50	1.24	....	....	1.96	18.90	....	902,000	23,900
21. " "	35.92	49.58	0.40	0.40	....	0.80	12.90	....	5.60	61.95	0.88	1.37	....	2.00	28.20	....	892,370	23,620
22. " "	22.00	67.00	0.60	0.60	3.00	0.80	6.00	....	3.40	83.20	1.30	2.10	6.60	2.10	1.30	....	895,050	22,030
23. Caspian region, Russia	....	92.24	....	3.50	....	....	4.26	....	....	84.52	....	8.60	....	....	6.88	....	1,062,000	21,570
24. " " "	....	97.54	....	2.47	....	....	....	....	....	93.35	....	6.65	....	....	....	....	1,050,000	22,200
25. " " "	....	95.56	....	4.44	....	....	....	....	....	88.00	....	12.00	....	....	....	....	1,030,000	20,900
26. Asperon Peninsula, Russia	0.34	92.49	....	0.93	2.13	....	4.11	....	0.04	87.38	....	2.43	3.35	...	6.80	....	1,061,670	22,280
27. " " "	0.98	93.09	....	2.18	0.40	....	3.26	....	0.16	88.14	....	5.55	0.83	...	5.32	....	1,063,360	22,170
28. Pechelbronn, Germany	....	77.03	3.50	3.60	8.90	1.80	4.80	....	....	64.00	5.00	8.20	12.90	3.00	6.90	....	908,200	16,000

* The calorific values given in the table are approximate only, as gases of the series $C_nH_n + 2$ are combined under the heading "Heavy Hydrocarbons." In making the calculations the calorific value of ethylene has been assumed as equal to that of the heavy hydrocarbons, since the bulk of the later consists of ethylene. For method of computing calorific value of natural gas, see page 133. For method of burning natural gas, see page 155.

hours per day the plant operates, and the quantity of coal needed for banking fires, the possibility of operating an oil fired plant where a coal fired plant would be objectionable owing to smoke, and many other similar considerations, far more than on the relative calorific value of oil and coal. In consequence there is but one reliable method of determining the relative advantages of the two fuels, and that is by operating the plant with each fuel for an interval of time long enough to give accurate data regarding costs of every item entering into the problem. Any other method must necessarily be approximate to such a degree as to render it practically guesswork.

Coal Tar usually has a value for other purposes far exceeding its fuel value, yet at times it is used to advantage for fuel. It differs from crude oil chemically, being lower in hydrogen and higher in carbon, and therefore of a lower calorific value. The following is an ultimate analysis of a tar made from a standard gas coal:

Carbon	89.21%
Hydrogen	4.95
Nitrogen	1.05
Oxygen	4.20
Ash	.06
Sulphur	.53
B. T. U. per pound	15,388

Water-Gas Tar is lighter than coal tar, and is the residuum of gas oil. Its analysis is as follows:

Carbon	92.70%
Hydrogen	6.13
Nitrogen	0.11
Oxygen	0.68
Ash	.05
Sulphur	.33
B. T. U. per pound,	17,296

A gallon of coal tar weighs 10.33 lbs., and a gallon of water-gas tar 9.58 lbs. In actual tests the former has evaporated 11.91 lbs. of water per pound of fuel, and the latter 14.9 lbs., from and at 212°.

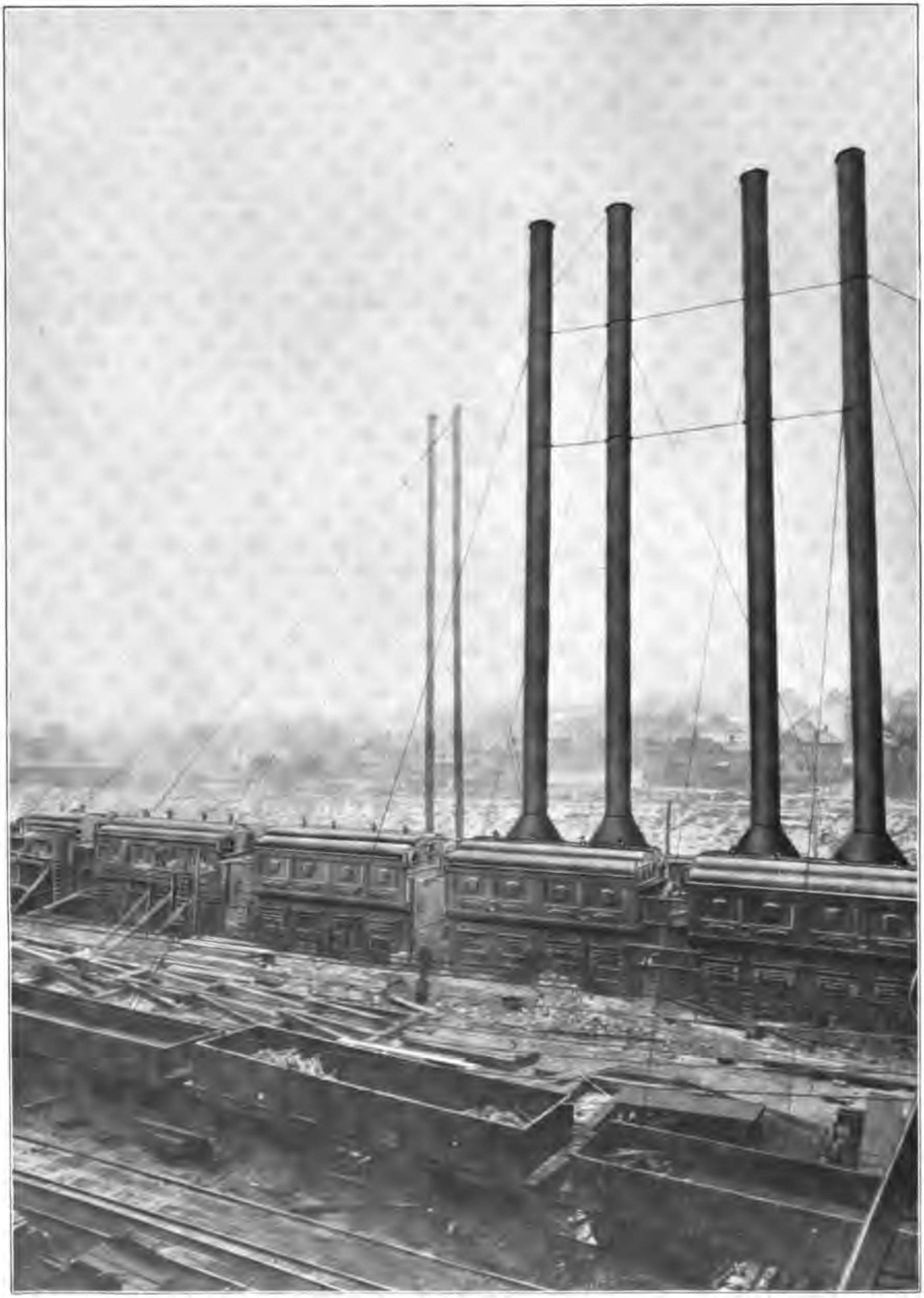
Natural Gas is pumped from the wells to the point where it is to be used. The gas leaves the pumping station in the field at pressures reaching 250 pounds per square inch; at the receiving station it is reduced to a pressure of 4 to 5 pounds before entering the distributing mains. At the boiler house this pressure is still further reduced by a valve controlled by the steam pressure. The final pressure at which the gas enters the burner is usually measured by a mercurial pressure gauge graduated to read in pounds and ounces per square inch. The charge for gas is based upon readings of a meter placed between the reducing valve and the burner. For purposes of comparison all observations should be based on gas reduced to standard temperature of 32° F. and absolute atmospheric pressure of 14.7 lbs. per square inch. When the temperature and pressure corresponding to the meter readings are known, the volume of gas under standard pressure and temperature can be obtained by multiplying the number of cubic feet indicated on the meter by $\frac{33.54P}{T}$ in which $P=$ *absolute* pressure in pounds per square inch, and T *absolute* temperature F. of the gas at the meter. In boiler tests the evaporation should be reduced to that per cubic foot of gas under standard pressure and temperature.

The weight of natural gas is about 45.6 lbs. per 1,000 cubic feet under standard conditions. The composition varies considerably, even in the same field. Table 45 gives analyses and calorific values of natural gases from various localities.

Comparison of Natural Gas and Coal—The same reasons which present any accurate comparison of the value of coal and petroleum without an actual test apply with equal force in case of coal and natural gas. The following table, based upon the assumption that one cubic foot of gas under standard conditions will evaporate .75 lb. of water, will enable an approximate comparison to be made.

WATER EVAPORATED PER POUND OF COAL	NO. OF M CU. FT. GAS EQUAL 2,000 LBS. COAL
7	18.7
8	21.3
9	24.0
10	26.7
11	29.3

Natural gas at 6 cents per 1,000 cubic feet will be equal in heating value to coal which evaporates 7 lbs. of water per pound and costs $1.12 per ton.



REPUBLIC IRON & STEEL CO. YOUNGSTOWN, O., OPERATING 11,300 H. P. OF STIRLING BOILERS

Determination of Heating Value of Fuels

Methods—The heating value of a fuel may be determined: (1) By calculation from a chemical analysis: (2) By burning a sample in a calorimeter. In the first method the calculation may be based on either an *ultimate analysis* or a *proximate analysis*. An ultimate analysis reduces the fuel to its elementary constituents of carbon, hydrogen, oxygen, nitrogen, sulphur, and the ash and moisture. The work requires the services of a chemist, and for further particulars the reader is referred to Stillman's *Engineering Chemistry*. A proximate analysis determines only the per cent. of fixed carbon, volatile matter, moisture, and ash, but does not determine the ultimate composition of the volatile matter.

Caution in Interpreting Results of Ultimate Analyses—Reports of ultimate analyses sometimes give the percentages of constituents referred to weight of the sample *less* its weight of moisture. When the report gives the proportions in this way and also the per cent. of moisture originally in the sample, the true analysis can easily be obtained, as shown in following case:

	CHEMIST'S REPORT	TRUE ANALYSIS
Carbon	76.91	72.25
Hydrogen	5.07	4.76
Oxygen	8.65	8.125
Nitrogen	1.16	1.09
Sulphur	1.21	1.135
Ash	7.00	6.58
	100.00	
Moisture	6.06	6.06
	106.06	100.00

The true analysis is obtained by dividing each of the apparent percentages, as reported, by the sum, 106.06.

The per cent. of moisture determined by drying a large sample, immediately after it is taken from the coal pile, will almost invariably be larger than determined from the analysis, because in shipping the sample to the chemist, and preparing it for analysis, some of the moisture evaporates.

The ultimate analysis resolves the fuel into its elementary constituents but does not reveal how these may have been combined in the fuel. The manner of their combination undoubtedly affects the calorific value, as fuels yielding identical ultimate analyses often give different heating values when tested in a calorimeter. The difference is very slight, and a very close approximation to the heating value may be computed from the ultimate analysis.

Calculations from an Ultimate Analysis—The first formula for the calculation of heating values from the composition of a fuel is due to Dulong, and this, slightly modified, is used to-day. Other formulas have been proposed, some of which give more accurate results for particular classes of fuels, but most of them are based upon Dulong's and are merely modifications of it. Dulong's formula* converted into British Units is

Heating value in B. T. U. per lb.=

$$14{,}500\,C + 62{,}100\left\{H - \frac{O}{8}\right\} \qquad [29]$$

The coefficients 14,500 and 62,100, representing the heat of combustion of carbon and hydrogen, have been investigated by numerous experimenters who determined values which differ slightly from those above given. With a view of establishing some uniform practise the American Society of Mechanical Engineers, in their "Rules for Conducting Boiler Trials," Code of 1899, recommend the following;

Heating value in B. T. U. per lb.=

$$14{,}600\,C + 62{,}000\left\{H - \frac{O}{8}\right\} + 4{,}000\,S \qquad [24]$$

*Dulong's original formula in French units is

$$\text{Heat Value in Calories} = 8080\,C + 34{,}500\left\{H - \frac{O}{8}\right\} \qquad [30]$$

The *calorie* is the heat unit of the metric system, and when used as a measure of the heating value of fuel, it is the number of *units* of weight of water which may be heated one degree Centigrade by the combustion of one unit weight of coal. The unit of weight may be either a kilogram, gram or pound. When thus used a *calorie* is equivalent to 1.8 B. T. U.

C, H, O, and S are the *proportional content* of carbon, hydrogen, oxygen, and sulphur, respectively. This formula is generally accepted by all American Engineers. The last term represents the heating value of formula gives results nearly identical with those obtained from calorimetric tests, and may safely be applied to all solid fuels except cannel coal, lignite, turf and wood, whenever a correct ultimate analysis is available.

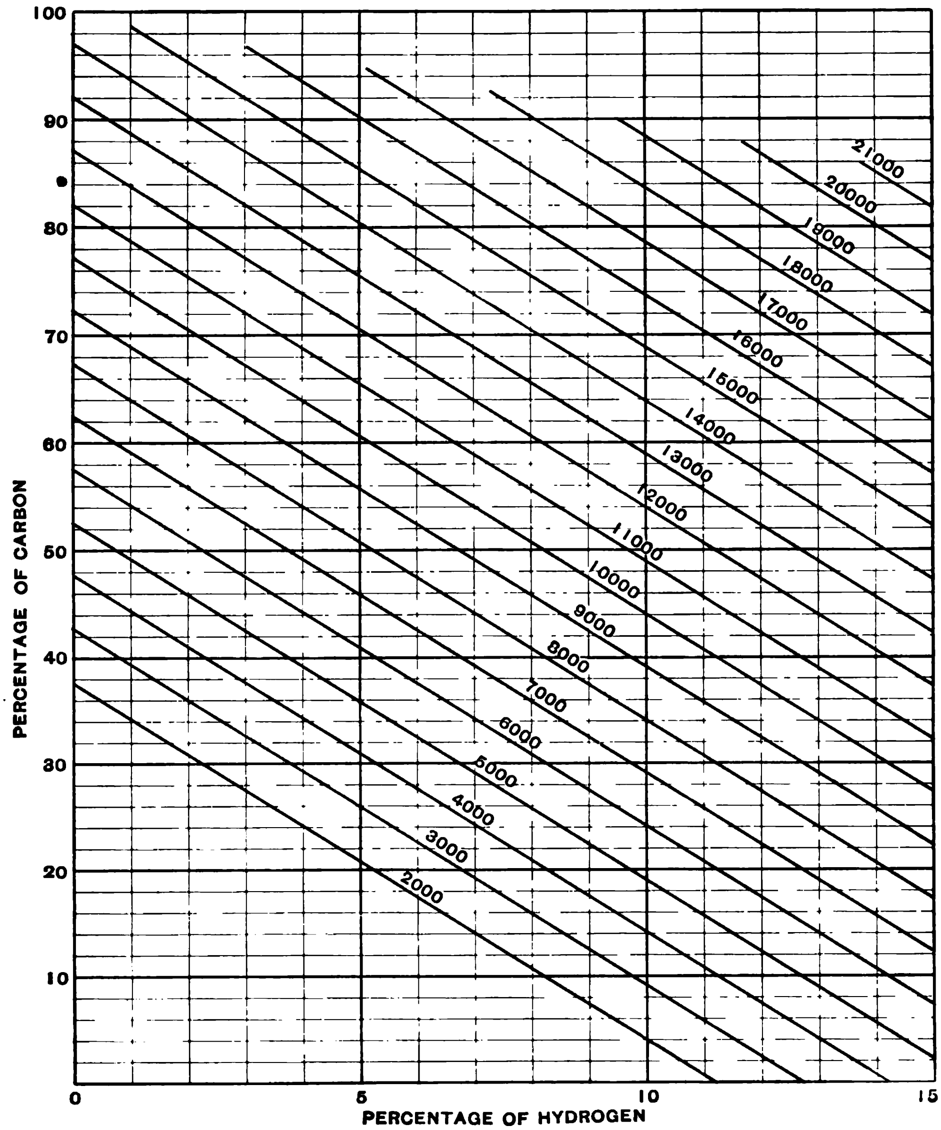


FIG. 25. CHART ILLUSTRATING MAHLER'S FUEL FORMULA

sulphur, based on determinations made by Lord. The method of using the formula has already been shown on page 106.

The investigations of Mahler in France, Lord and Haas in this country, and Bunte in Germany, all show that the Dulong

Mahler's Formula* is based upon the content of carbon and hydrogen only. It is simpler than Dulong's, and sufficiently accurate for many practical purposes. It is:

$$\text{B. T. U. per pound of fuel} = 201\,C + 676\,H - 5540 \qquad [31]$$

*For derivation of this formula see *The Locomotive*, November, 1903.

in which C and H are the *percentages* by weight, of carbon and hydrogen in the fuel. An advantage of this formula is that the results may, without calculation, be obtained from the diagram, Fig. 25. Example: To find the calorific value of a fuel containing four per cent. hydrogen, eighty-four per cent. carbon, and twelve per cent. of ash, water, etc., locate on the hydrogen scale at the bottom the line under four; pass vertically upward along this line until it intersects the horizontal line passing through eighty-four on the carbon scale. The point of intersection is on the line marked 14,000, hence the fuel contains 14,000 B. T. U. per pound.

Heat Values of Gaseous Fuels—The method of computing calorific values from ultimate analyses is particularly adapted to solid fuels, with the exceptions already noted. In the case of gaseous fuels, it is better to separate them into their elementary constituent gases, and to compute the heating values of these gases separately. Usually only hydrogen, carbon monoxide (CO) and certain hydrocarbons will be found constituting the combustible portion of the gas. Table 47 gives calorific value of the commonest combustible gases.

Application of the table. As gas analyses may be reported either by weight or by volume, an example of each will be given:

TABLE 47

WEIGHTS AND CALORIFIC VALUE OF GASES AT 32° F. AND ATMOSPHERIC PRESSURE

GAS.	Chemical Symbol	Cubic Feet per Pound of Gas.	B. T. U. per Pound of Gas.	Cu. Ft. of Air* Required per Pound of Gas.	B. T. U. per Cubic Foot of Gas.	Cu. Ft. of Air Required per Cubic Foot of Gas.
Hydrogen	H	178.93	62,000	428.25	346	2.39
Carbon Monoxide	CO	12.81	4,350	30.60	339	2.39
Marsh Gas	CH_4	22.43	23,564	214.00	1050	9.54
Acetylene	C_2H_2	13.79	21,465	164.87	1556	11.93
Olefiant Gas	C_2H_4	12.80	21,440	183.60	1675	14.33
Ethane	C_2H_6	11.96	22,230	199.88	1859	16.72

*To reduce volumes of air to pounds of air multiply by 12.39.

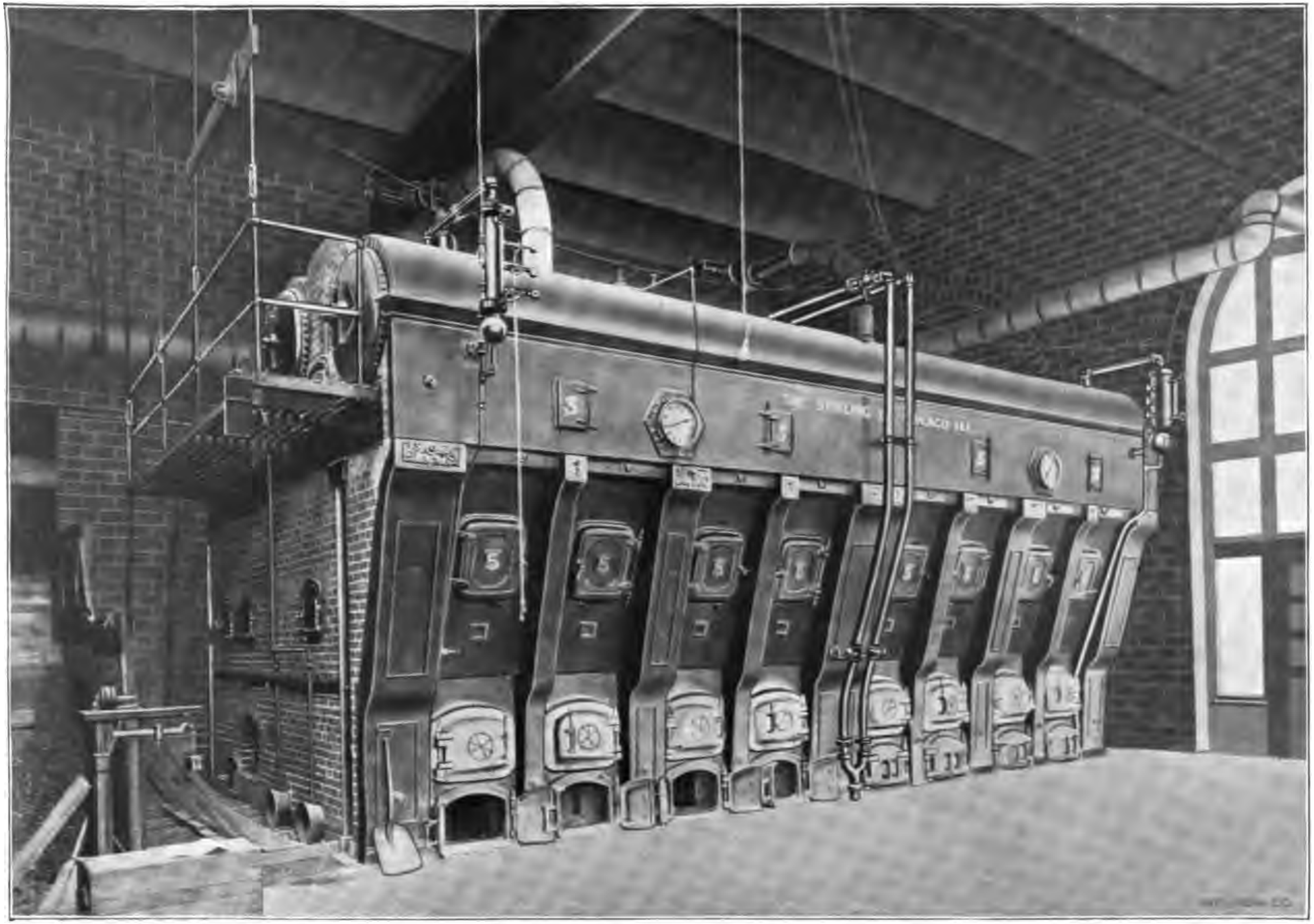
Correction for Hydrogen, Moisture and Nitrogen—If the fuel contains water, the heat necessary to evaporate it and to superheat the steam thus formed produces no useful result and should be deducted from the amount given by the above formulas. The same thing applies to the water formed from the hydrogen present. The nitrogen in the fuel absorbs heat without producing any benefit. The total losses due to these causes can be computed by the formula

$$\text{B. T. U. Lost}= (9H+W)[212.9-t+965.8+0.48(t_c-212)] + 0.2438(t_c-t)N \quad [32]$$

In which H, W, and N are the proportional content of hydrogen, water and nitrogen, t_c the temperature of breeching, and t the temperature of air supply.

(1) A blast furnace gas, analysis by weight being, oxygen (O) = 2.7; carbon monoxide (CO) = 19.5; carbon dioxide (CO_2) = 18.7; nitrogen (N) = 59.1; all in per cents. The only combustible present is carbon monoxide, hence the heating value per pound of the gas is 0.195×4350=848.25 B. T. U. The *net* volume of air needed to burn a pound of the gas is 0.195×30.6=5.967 cu. ft.

(2) A natural gas, analysis by volume being, oxygen (O)=0.40; carbon monoxide (CO)=0.95; carbon dioxide (CO_2)=0.34; olefiant gas (C_2H_4)=0.66; ethane (C_2H_6)=3.55; marsh gas (CH_4)=72.15; hydrogen (H)=21.95, all in per cents. All but the O and CO_2 are combustibles, hence the heat developed and net air required per pound of gas will be as worked out in detail in the following table:



600 H. P. OF STIRLING BOILERS, PHILADELPHIA MUSEUM'S EXPOSITION

Heat from	CO	$=0.0095\times339=$	3.22	B. T. U.
" "	C_2H	$=0.0066\times1675=$	11.05	" " "
" "	CH	$=0.0355\times1859=$	65.99	" " "
" "	CH	$=0.7215\times1050=$	757.58	" " "
" "	H	$=0.2195\times346=$	75.95	" " "
		Total.	913.79	B. T. U.

Air needed for	CO	$=0.0095\times2.39=$	0.022705	Cu. ft.
" " "	C_2H_4	$=0.0066\times14.33=$	0.094578	Cu. ft.
" " "	C_2H_6	$=0.0355\times16.72=$	0.593560	Cu. ft.
" " "	CH_4	$=0.7215\times9.54=$	6.883110	Cu. ft.
" " "	H	$=0.2195\times2.39=$	0.524605	Cu. ft.
		Total air needed,	8.118558	Cu. ft.

Proximate Analysis—The proximate analysis of fuel gives its proportions of fixed carbon, volatile combustible matter, moisture and ash. It is made by subjecting a sample to a temperature of 250° to 300° to expel the moisture, then to a red heat which expels the volatile matter; then to a white heat which causes the carbon to pass off as dioxide, leaving the ash as a residue. By weighing the residue at end of each operation the various percentages can be computed. See Article XV of Code, in chapter on Rules for Conducting Boiler Trials, page 204.

Table 48 gives ultimate and proximate analyses of Alabama coals, and illustrates the relationship between the two.

The proximate analysis is easy to make, it affords information as to the general characteristics of a fuel, and its relative heating value, but from it the heating value cannot be directly computed. The reason is that the volatile content varies widely in composition and heating value.

Comparison of many experiments has resulted in production of some methods of estimating the calorific value of coals from proximate analyses. Kent* deduced from Mahler's tests on European coals the approximate heating values of coal dependent upon the content of fixed carbon in the combustible† as given in the following table.

TABLE 49

APPROXIMATE HEATING VALUE OF COALS

(*Kent.*)

Percentage Fixed Carbon in Coal Dry and Free from Ash.	Heating Value B. T. U. per Pound Combustible.	Percentage Fixed Carbon in Coal, Dry and Free from Ash.	Heating Value B. T. U. per Pound Combustible.
100	14,600	68	15,480
97	14,940	63	15,120
94	15,210	60	14,580
90	15,480	57	14,040
87	15,660	55	13,320
80	15,840	53	12,600
72	15,660	51	12,240

Example: Given a coal whose proximate analysis is, fixed carbon 61%, volatile matter 29%, ash 8%, moisture 2%. The combustible portion amounts to 61+29=90% of which the fixed carbon is 61÷90=68%. From Table 49 the *combustible* portion of such a coal has a heat value of 15,480 B. T. U.; hence the correct heating value, per pound of coal, is

$$15{,}480\times.90=13{,}932 \text{ B. T. U.}$$

TABLE 48

PROXIMATE AND ULTIMATE ANALYSES OF ALABAMA COALS

Name of Seam.	Location.	Proximate Analyses.		Ultimate Analyses.					Common to Proximate and Ultimate Analyses.	
		Volatile and Combustible Matter.	Fixed Carbon.	Carbon.	Hydrogen	Oxygen.	Nitrogen.	Sulphur.	Ash.	Moisture.
Wadsworth	Helene	34.30	60.50	73.23	7.98	11.92	1.07	0.60	3.50	1.70
Pratt	Pratt	33.45	63.20	75.82	10.52	7.51	1.73	1.07	2.00	1.35
Brookwood	Brookwood	27.80	58.70	72.47	10.38	1.60	0.40	1.65	11.90	1.60
Woodstock	Blocton	34.80	60.60	72.75	8.61	11.12	1.48	1.44	2.65	1.05
Underwood	Blocton	35.65	57.30	70.82	10.19	9.95	1.31	0.68	5.25	1.80
Pratt	Pratt	31.55	64.05	75.05	9.91	8.05	1.62	0.07	2.35	1.15
Milldale	Brookwood	30.50	66.30	73.96	10.50	9.57	1.62	1.15	2.20	1.00
	Blue Creek	25.80	69.00	72.68	10.77	9.83	1.39	1.03	2.80	1.50
	Coalburg	32.55	65.57	74.59	10.58	9.48	1.31	1.32	1.90	.82
Cahaba		30.15	52.00	60.37	10.70	9.00	1.26	1.72	16.30	.65

*Steam Boiler Economy, First Edition, p. 47.

†See foot-note, page 112.

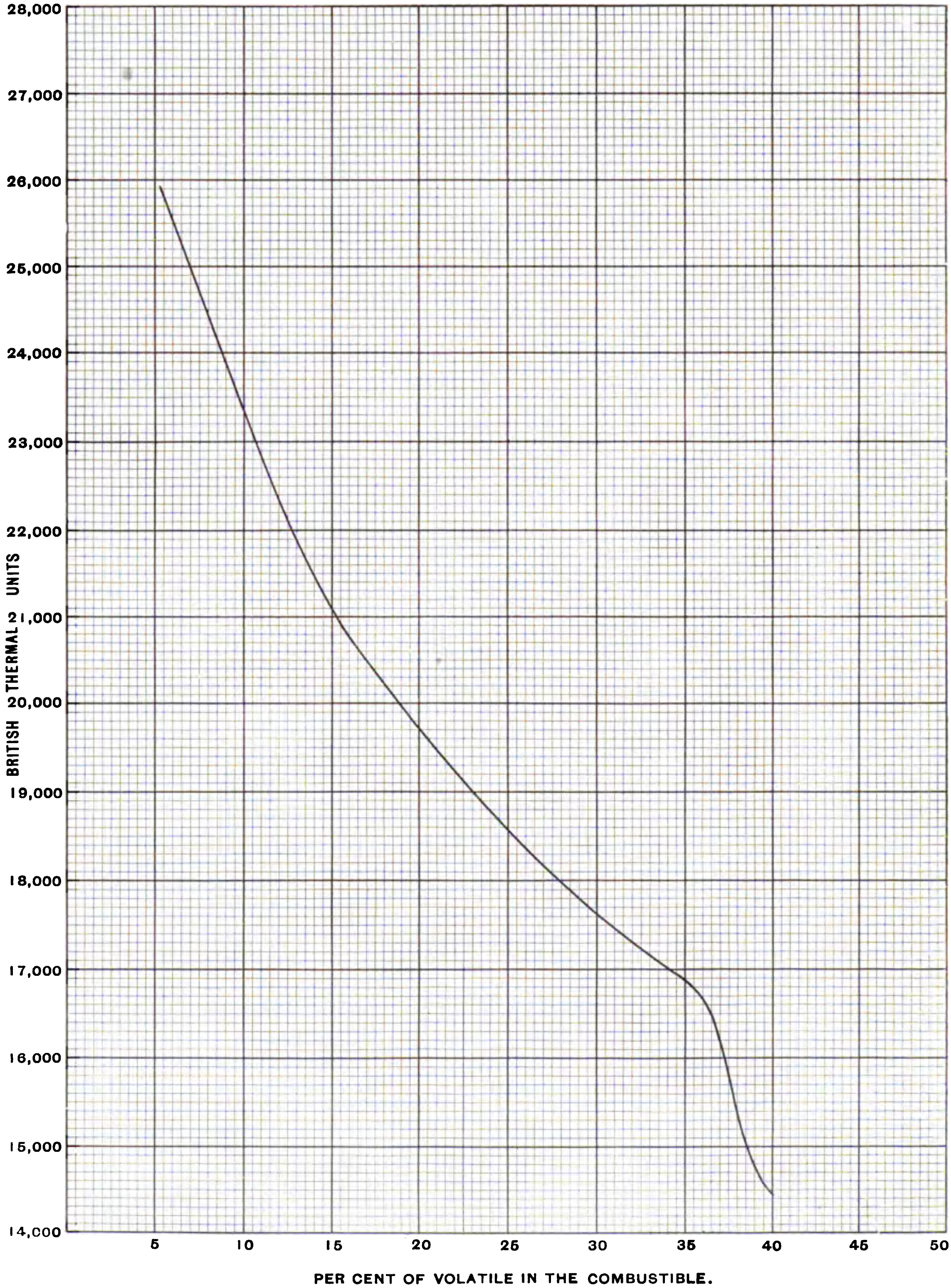


FIG. 26. GOUTAL'S VALUES FOR "A" IN B. T. U.=14,760 C+aV

To facilitate the use of Kent's method, Fig. 27 has been prepared; the per cent. of fixed carbon in the combustible having been located on the abscissa, the B. T. U. per pound of combustible can be determined from the corresponding ordinate.

Goutal* gives carbon a fixed value, and considers the heat value of the volatile matter a function of its percentage referred to combustible. Goutal's formula, in British units, is,

$$B.T.U.\ per\ lb.\ of\ coal = 14{,}760C + aV \quad [33]$$

In which

$C =$ the proportional content of fixed carbon in the *coal*.

$V =$ the proportional content of volatile matter in the *coal*.

$a =$ a variable depending on the ratio V′ of volatile matter to *combustible*, per following table, or from Fig. 26.

$V' = \frac{V}{V+C}$	a
.05	26,100
.10	23,400
.15	21,060
.20	19,620
.25	18,540
.30	17,640
.35	16,920
.38	15,300
.40	14,400

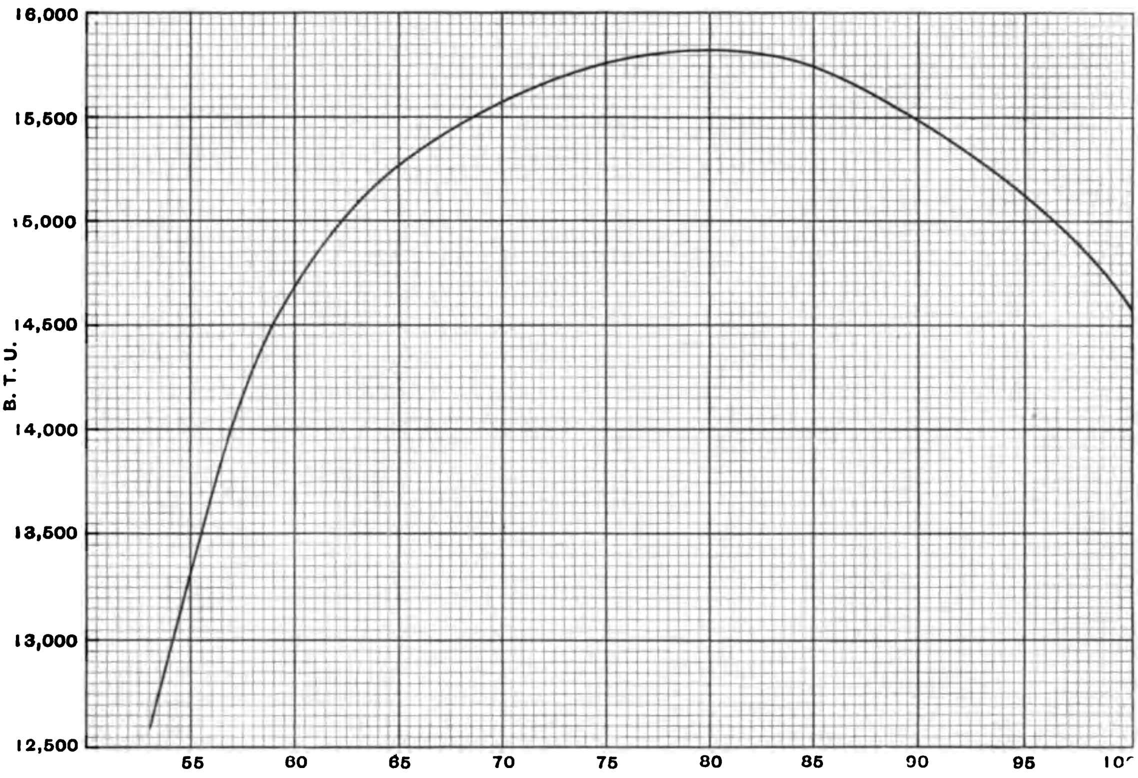


FIG. 27. GRAPHICAL REPRESENTATION OF THE RELATION BETWEEN PERCENTAGE OF FIXED CARBON IN COMBUSTIBLE, AND THE CALORIFIC VALUE PER POUND OF COMBUSTIBLE

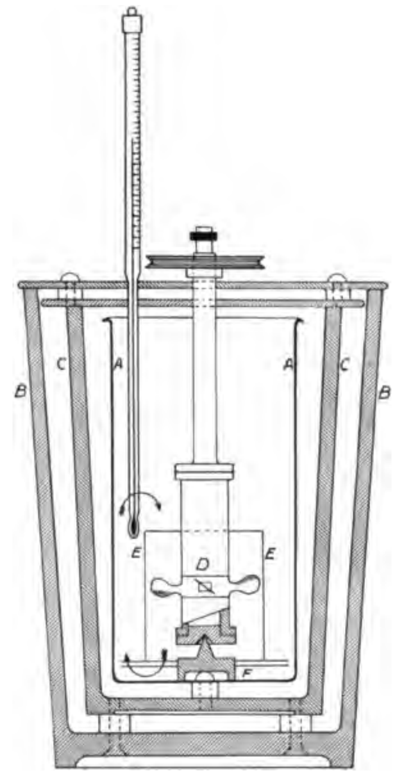
Applying the formula to the same coal as in preceding example, $C = 0.61$; $V = 0.29$; $V' = \frac{0.29}{.61 + .29} = .32$, hence from the figure "a" = 17,300, hence B. T. U. per pound of coal = $14{,}760 \times 0.61 + 17{,}300 \times 0.29 = 14{,}020$, which is only about six-tenths of one per cent. different from the value found by Kent's method.

Comptes rendus de l'Academie des Sciences, Vol. cxxxv, p. 477.

Illinois Coals — From calorimetric determinations and chemical analyses of over a thousand samples of coal, R. W. Hunt & Co., deduced the formula,

$$\text{B. T. U. per pound of coal} = 14{,}544C + 16{,}515V - 10{,}000A \qquad [34]$$

which is correct within narrow limits for Illinois coals in which the content of fixed carbon and volatile matter ranges from 40 to 45 per cent.; when the ash lies between 10 to 15



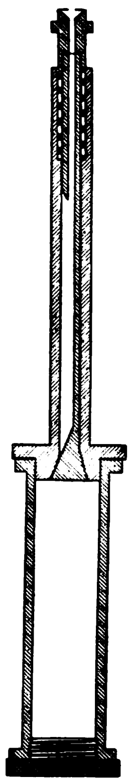


FIG. 28 FIG. 29

PARR'S FUEL CALORIMETER

per cent. the formula will be more accurate if written.

$$\text{B. T. U. per pound of coal} = 14{,}544C + 16{,}515V + 354A - 1635 \qquad [35]$$

In both cases C, V, and A are the proportional content of fixed carbon, volatile matter. and ash.

Range of Accuracy of Fuel Formulas—Mr. Kent states that for coals containing sixty per cent. or more of fixed carbon in the combustible, the values in Table 49 are practically correct, but for coals containing less than sixty per cent. of fixed carbon the tabular values are liable to an error of four per cent. in either direction.

M. Goutal states that his formula proved very accurate over a wide range of experiments, six hundred different coals being used, and that the error rarely exceeded one per cent.; it was found to give values two per cent. high for some anthracites and lignites.

So far as the present writer has been able to test these two methods, they give results which are accurate enough for all ordinary work, when applied to eastern coals whose percentage of fixed carbon and volatile matter fall within their range, but they apply with less accuracy in proportion as the coals are mined in the fields farther to the west, and for fuels mined in Wyoming, Colorado and farther west and north the formulas are of little use. Consequently, while fuel formulas are of great value where approximate results only are necessary, a calorimetric determination of the heating value of the fuel is necessary whenever exact results are required.

Calorimetry—The ultimate or proximate analysis of a fuel is useful in determining its general character, and in making a close approximation to its heating value; but for a practical determination of heating value the calorimeter method is more satisfactory. In this a sample of the fuel is actually burned, and the heat of combustion is measured.

Calorimeters are composed of a combustion chamber and a calorimeter bath, the latter consisting of a vessel surrounding the combustion chamber, and containing a known quantity of water. The elevation of the temperature of the water, when accurately measured and multiplied by suitable constants peculiar to the apparatus, determines the heating power of the fuel.

Mahler's Calorimeter is very popular and much used, but its operation is very complicated, and requires an expert. Both the instrument and method of operating it are described in Kent's *Steam Boiler Economy.*

Parr Calorimeter—A very reliable, inexpensive, and simple calorimeter is that invented by Prof. S. W. Parr, of the University of Illinois. This apparatus does not require

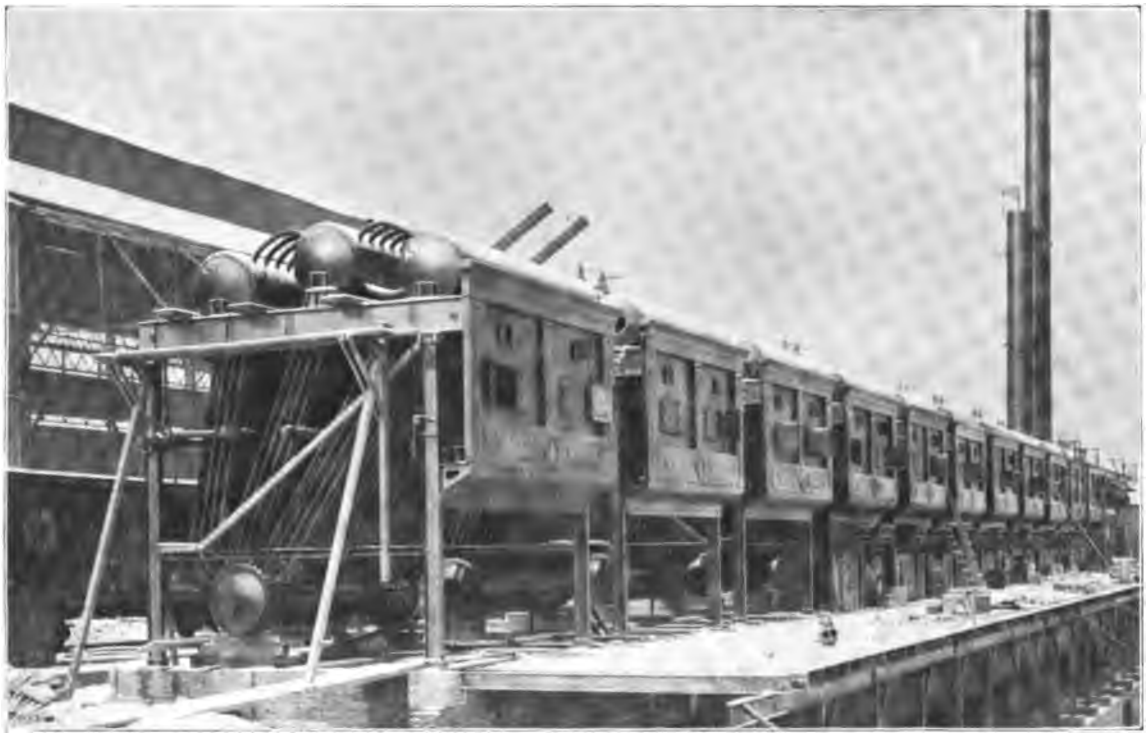
the services of an expert operator. Oxygen is not used, no high pressures are employed, and the total time consumed in conducting a test on a weighed and dried sample should not exceed 15 or 20 minutes.

Fig. 28 shows the relative position of parts. The can *A* is filled with two litres of water. The combustion takes place within the cartridge *D*. The resulting heat is imparted to the water. The rise in temperature is indicated by the finely graduated thermometer *T*. Fig. 29 shows the cartridge in which is placed a weighed quantity of coal, previously ground to pass through a 100 mesh sieve and dried in the usual way at 220° to 230° F. There is also put into the cartridge a chemical compound which is thoroughly mixed with the coal by shaking. The cartridge is then placed into a measured quantity of water in the insulated calorimeter can *A*. The stirrer is set in motion and operated by a cord about the pulley *P*. After a constant temperature has been attained ignition is effected by means of a short piece of hot wire dropped through the stem of the cartridge.

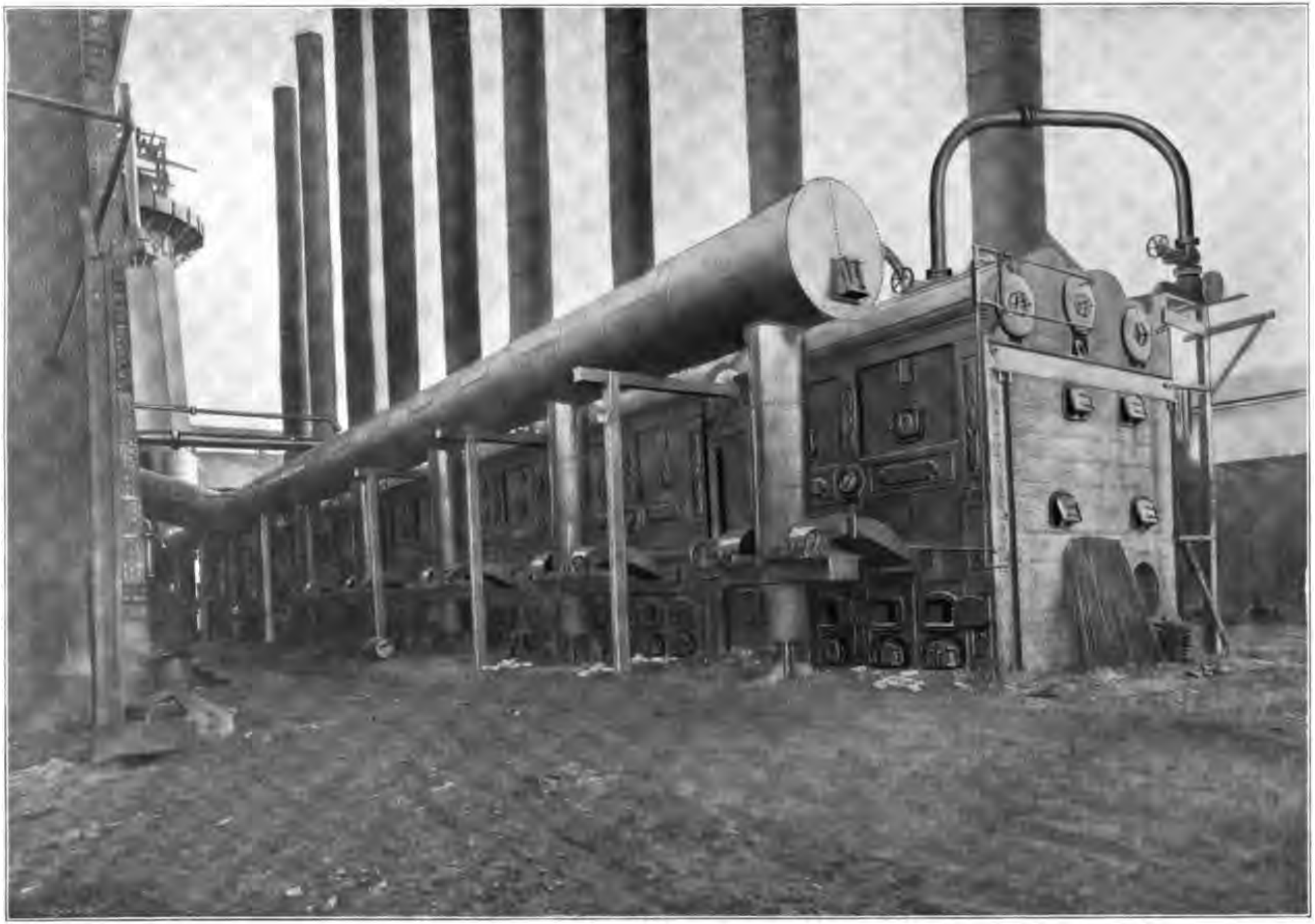
Extraction of the heat is complete in from four to five minutes. The maximum reading is taken and the rise in temperature, multiplied by a simple factor, gives the heat in British thermal units per pound of coal. By a slight modification of the apparatus, ignition may also be effected by an electric fuse, and where current is available this method is preferred by some users.

The instrument is well adapted to the determination of sulphur in coal, pyrites, petroleum, etc. Upon dissolving out the products of combustion from the bomb the sulphur of the original material, being in the form of soluble sulphate, may very readily be made to indicate the percentage content by a simple photometric device.

The residue from the combustion contains the carbon of the coal in the form of sodium carbonate. The volume of carbon dioxide may readily be measured, and from this the total carbon of the coal can be calculated. This is a result not heretofore available except by ultimate analysis, and enhances the value of the instrument.



ST. CLAIR STEEL COMPANY, CLAIRTON, PA., OPERATING 6,500 H. P. OF STIRLING BOILERS



ALABAMA STEEL AND WIRE CO., BIRMINGHAM, ALA., OPERATING 11,600 H. P. OF STIRLING BOILERS

Fuel Burning

The preceding chapter indicates the wide range of the nature and calorific value of the available boiler fuels; the methods of burning these fuels to best advantage will now receive attention.

Draft—The intensity of draft required varies with the kind and amount of fuel to be burned per square foot of grate, as shown by Fig. 38* in the chapter on Chimneys. It is well known that if the draft is deficient, the volatile matter in the fuel escapes unburnt with the furnace gases, and the fire is dead and smoky. It is not generally recognized that an excess of draft causes equally large losses by burning holes through the fire and admitting surplus air which reduces the furnace temperature. Consequently, to secure the most efficient results the draft should be regulated by the damper to just the amount corresponding to the desired combustion rate, and *no more.*

Anthracite may be burned in almost any kind of furnace, but the grate area, and the intensity of draft must be sufficient to burn the amount of coal requisite to develop the desired capacity. When possible the coal should be at least 6 inches deep on the grates, because with thinner fires air holes are liable to form in the bed of coals. The smaller sizes of anthracite require more draft than the larger sizes, and the light weight of the coal particles renders it difficult to prevent the draft forcing holes through the fuel. If a thick fire is maintained so as to avoid an excess of air, the tendency of the fuel is to choke the interstices in the grate-bars and to cause a deficit of air. To keep between these limits and obtain just the correct amount of air requires considerable skill on the part of the fireman. The fires require frequent cleaning, and as the size of the coal decreases there is likely to be trouble from clinkers.

The successful burning of these small sizes requires a grate with a large number of very small air openings, and usually forced draft. When the coal clinkers a steam jet blowing into the ash pit will be found beneficial. Shaking grates may also be used to advantage, since they make it possible to rid the fires of ash without disturbing them to any great extent on the surface. *Once anthracite is placed into the furnace it should not again be touched except when it is necessary to clean the fire.*

In proportion as the coal is coarser, more of it may be fired at each charge. The proper interval between charges can be determined by careful observation of the fire. After the fire reaches a white heat the lower part of the bed of coals will burn away, and the upper surface will sink; this sinking indicates the proper moment for firing again because unless fresh coal is quickly added, air holes will form in the fire. In one or two minutes after the new charge of coal is fired, flame will appear over this coal in in spots which indicate uneven flow of air through the fuel. These spots should immediately be covered with additional fresh coal so spread as to compel the air to pass at a uniform rate through the entire bed of fuel. No further coal should be thrown upon the fire until it sinks again, otherwise the formation of clinkers will be considerably increased.

The Stirling furnace is perfectly adapted to anthracite, and the incandescent arch supplies the heat necessary to ignite the carbon monoxide distilled from the fresh coal.

Volatile Matter—All coals except anthracite contain a considerable portion of volatile matter which must be burned to develop the full heating power of the fuel. How to do this has always been a most troublesome problem which is seldom solved in boiler furnaces. The per cent. of volatile matter steadily increases in the progression from anthracite to lignite; accordingly, as the coal is of poorer grade not only is its calorific power less, but it becomes more difficult to develop what it has. The reason for this lies principally in the failure to adapt the furnace to the peculiarities of the coal.

When fresh bituminous coal is thrown upon the fire "the first thing that the fine fresh coal does is to choke the air spaces existing through the bed of coke, thus shutting off the air supply which is needed

*See page 174.

to burn the gases produced from the fresh coal. The next thing is a very rapid evaporation of moisture from the coal, a chilling process, which robs the furnace of heat. Next is the formation of water-gas by the chemical reaction, $C+H_2O=CO+2H$, the steam being decomposed, its oxygen burning the carbon of the coal to carbonic oxide, and the hydrogen being liberated. This reaction takes place when steam is brought in contact with highly heated carbon. This also is a chilling process, absorbing heat from the furnace. The two valuable fuel-gases thus generated would give back all the heat absorbed in their formation if they could be burned, but there is not enough air in the furnace to burn them. Admitting extra air through the fire-door at this time will be of no service, for the gases being comparatively cool cannot be burned unless the air is highly heated. After all the moisture has been driven off from the coal, the distillation of hydrocarbons begins, and a considerable portion of them escapes unburned, owing to the deficiency of hot air, and to their being chilled by the relatively cool heating surfaces of the boiler. During all this time great volumes of smoke are escaping from the chimney, together with unburned hydrogen, hydrocarbons, and carbonic oxide, all fuel-gases, while at the same time soot is being deposited on the heating surface, diminishing its efficiency in transmitting heat to water.''*

To burn these gases it is necessary that they be brought into contact with a supply of air hot enough to cause ignition, and that they have ample space in which to mix with the air and burn completely before coming into contact with the boiler surfaces which are comparatively cool and extinguish the flame.

Inefficient Furnaces—Few boiler furnaces comply with these requirements. In the internally-fired boiler the furnace is surrounded with water so that the gases are liberated in a space which is too restricted to permit proper mixture with air, and too cold to cause ignition. In the return tubular boiler there is more space available for mixing the gases and air but the flame is extinguished by the cool boiler shell which forms the top of the furnace. In the horizontal water-tube boiler the roof of the furnace is a nest of water-tubes, and any flame not extinguished by first contact with them is extinguished by being drawn between them and surrounded by water-cooled surfaces. Complete combustion of volatile matter in such boiler furnaces is therefore impossible.

The Stirling Furnace has already been described, and its adaptation to burning volatile matter set forth.† An abundant supply of air can be admitted to the gases at all times, and as the furnace is surrounded by incandescent fire-brick the heat necessary for complete ignition of the gases is available and *at the right place.* The distinguishing feature of the Stirling furnace is the fire-arch. That this is an indispensable part of any furnace efficient for burning volatile matter is recognized by engineers. Many opinions in support of this statement might be quoted, but the following must here suffice.

"Chilling the gases before combustion is complete, should be carefully prevented; and comparatively cold surfaces, as those of a steam boiler, should not be placed too near the burning fuel. A large combustion chamber should, where possible, be provided, and more complete combustion may be expected in furnaces of large size, *lined with fire-brick, and with arches of the same material*, than in a furnace of small size where the fire is surrounded by chilling surfaces, as in a 'fire-box steam boiler'." (R. H. Thurston, *A Manual of Steam Boilers*, 7th ed., p. 188.)

"The change required in the furnace is the *roofing of it with fire-brick.......*" (Kent, *Steam Boiler Economy*, 1901, p. 159.)

"Of all the different kinds of furnaces designed for various purposes, the most persistent smoker is that of the steam boiler. The reason is obvious, for there are not hot walls to radiate back the heat and thus aid combustion. In some designs of boilers the furnace is *enclosed in a fire-brick combustion chamber*, and the products are not admitted to the heating surfaces until after combustion has become more or less perfect. This arrangement has met with success in many instances ...." (H. De B. Parsons, *Steam Boilers*, 1903, p. 15.)

Bituminous Coals and Lignites—The difficulties encountered in burning bituminous coal with economy and without

*Kent; *Steam Boiler Economy*, p. 155. †See pages 10 and 11.

smoke increase as the content of fixed carbon grows less; the coals requiring the greatest skill in handling are those of the bituminous variety from Illinois, Iowa, Missouri and the West. To burn the volatile matter the furnace must be large, to permit the air and gases to mingle; hot, to ignite the mixture and complete the combustion before he boiler surface is reached; and provided with ample grate surface to burn the requisite quantity of fuel—requirements prefectly met in the Stirling furnace.

The fire needs more attention than in case of anthracite. The fixed carbon will usually take care of itself if the fire is so handled as to burn the volatile matter. The depth of coal to be carried on the grates to produce the best results varies through wide limits according to the nature of the coal. Coa's from the same locality may require different depths, hence it is impossible to give any general rule applicable to all cases. The fireman must, by careful trials with each coal, determine the proper depth; the following information may serve as a suggestion when making such trials.

Semi-bitumious coals, such as Pocahontas, New River, Clearfield, etc., require fires from 12 to 14 inches thick; fresh coal should be charged at intervals of 10 to 20 minutes, and the quantity should be sufficient to maintain the thickness above given. Bituminous coals from the Pittsburg district require fires 4 to 6 inches deep, and should be fired often and in comparatively small charges. The coals mined in Kentucky, Tennessee, Ohio, and Illinois require a depth of 3 to 4 inches. Free-burning coals from Rock Springs, Wyoming, require 6 to 8 inches, while the poorer coals of Montana, Utah and Washington require a depth of about 4 inches. Colorado lignites require a depth of 4 to 6 inches, and grates with air spaces only $\frac{1}{4}$ to $\frac{5}{16}$-inch wide. Nova Scotia coals require a large supply of air, and the bed of coals must be so thin as barely to cover the grates.

In general, the coals mined in the western part of the United States require thinner fires than the eastern coals. If thicker fires are carried the tendency to clinker is increased. When burning these, fresh fuel should be fired often and in small amounts.

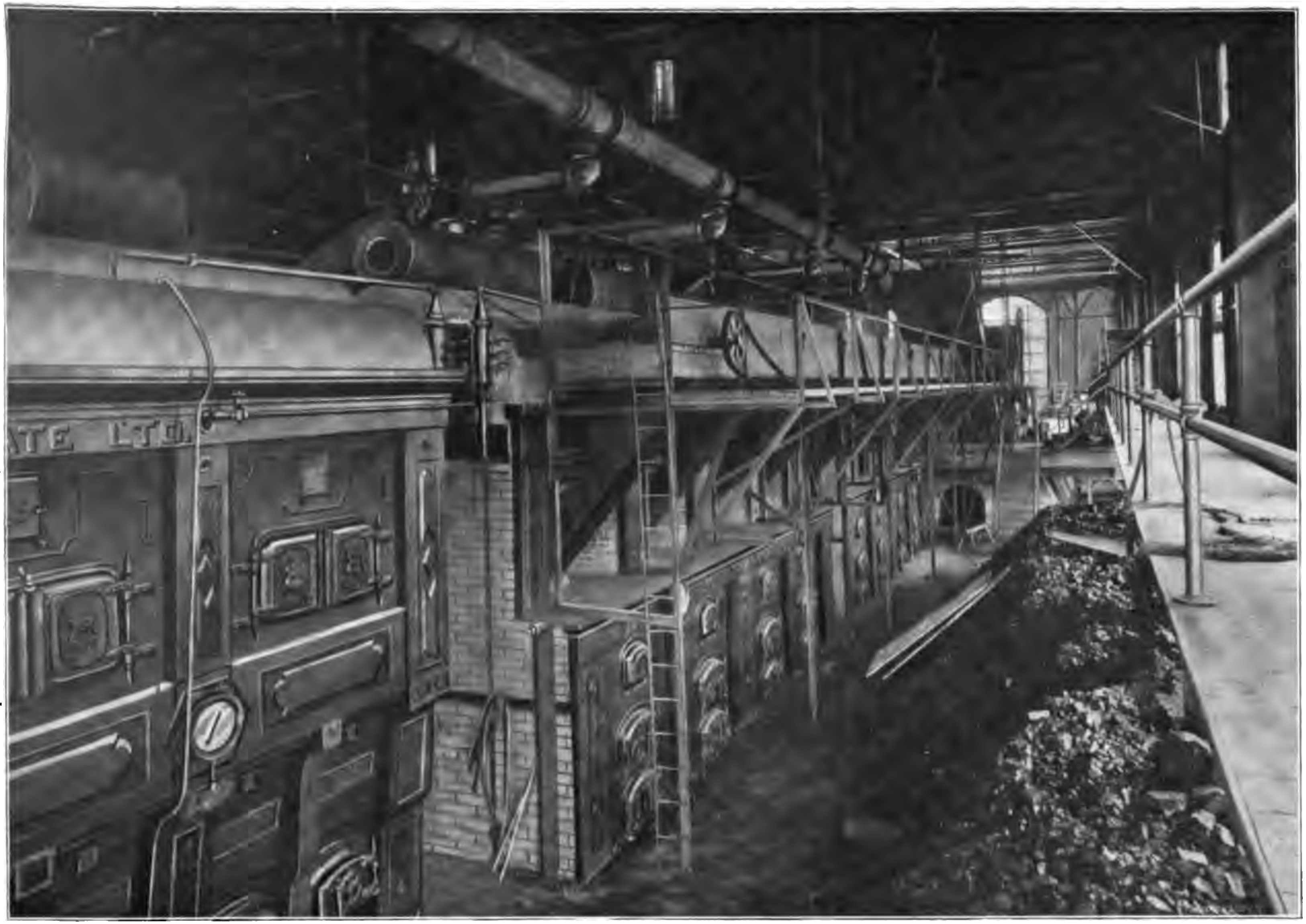
With hand firing there are three methods of feeding the coal:

(1) **The Alternate**, in which the fresh coal is fired on one side of the grate at a time. The volatile matter distilled from the fresh charge can be effectively burned by the air which is heated when passing through the other side; thus the two important stages of coal burning are made to occur at once,—the combustion of the volatile matter, and the burning of carbon. This obviates the necessity of continually altering the air supply to correspond first with one stage, and then the other. The alternate method gives excellent results when properly carried out.

In this method, and in the spread-firing method next to be described, the coal should be thrown exactly where it is wanted, and not be further disturbed by poker or slice bar, except when absolutely necessary to clean fires or break up clinkers.

(2) **In Spread-firing** very little fuel is charged at one time, and this is either deftly spread over the entire fuel bed, or in patches. Firing "lightly and often" is spread-firing practically. Where the fuel is laid in patches some of the advantages of the alternate method are obtained, but it has the disadvantage that the entire grate must be cleaned at one time. This method is fairly successful with small sizes of free-burning coals in furnaces where the gases rise vertically.

(3) **The Coking Method** consists of firing the fresh coal to a considerable depth directly in front of the firing doors, and pushing it back into the furnace as soon as it has coked. This results in a very hot fire in the rear of the furnace, due to burning carbon, and if the volatile matter from the fresh coal passes over this highly heated portion the combustion will be perfect provided the air supply is correct. This method is not particularly successful where the volatile matter rises vertically in the boiler, as is the case in horizontal water-tube boilers employing vertical baffles. In the Stirling the arched furnace directs the gases *horizontally* for a considerable distance, and the coking method has given very satisfactory results with coals containing a large amount of volatile matter. This method of firing was once very extensively employed, but is now going out of use. A



1,800 H. P. OF STIRLING BOILERS, WITH BAGASSE FURNACES, KENILWORTH SUGAR ESTATE, LT'D, KENILWORTH, LA.

disadvantage is that air passes much more easily through the coked coal than through the fresh coal, hence it is necessary to maintain a considerably greater depth of fuel on the rear of the grates than on the front. The fuel is also stirred up when it is pushed toward the rear of the grates. and it is now recognized that the less the coal is disturbed after it is fired, the more efficiently it can be burned. To use the coking method successfully the fireman must not only possess considerable skill, but must also give his undivided attention to the work.

The best method to adopt will depend upon the character of the fuel, and like all other work done by manual labor, on the "personal equation" of the fireman, but there should be *some* method followed, and the firing not be done haphazardly. A careful trial of the three methods will show which one is best adapted to the conditions, and that one should be adhered to. There may be a difference of from 10 to 20 per cent., between the results obtained from careless and skilled firing. Few boiler owners realize the saving to be effected by employing skilled and conscientious firemen.

Mechanical Stokers—Of these there are two general classes:

(*a*) Over-feed. (*b*) Under-feed.

The first spreads the fresh coal over the fuel bed, and the second feeds it below the grates, then upward, until it overflows out over the grates.

There are three kinds of over-feed stokers in use. In one the coal is carried on horizontal or slightly inclined grate bars, and the individual bars are given a mechanical motion by which the coal is gradually advanced along the grates toward the bridge wall. In another the grates are steeply inclined, and the fuel is pushed on to the upper ends, whence it slides down slowly toward the ash-pit, burning in transit. The third kind includes "chain grates," in which the entire grate is an endless chain of short bars. The motion is from the fuel hopper in front of the boiler, back toward the bridge wall, at which point the grate passes over a sprocket, then returns through the ash pit. The Stirling Chain Grate is typical of this class.

Underfeed stokers feed into a receptacle below the grates, and the fuel gradually overflows out onto the grates. It undergoes a coking process in the receptacle, and should be free from all volatile matter when the grates are reached. Some of these stokers operate intermittently, by means of a plunger; others feed continuously through a screw motion and forced draft is used.

In favor of mechanical stokers it is urged that they reduce the cost of fire-room labor, cause a slightly increased evaporation per pound of coal, permit of the use of coal-conveying apparatus, and lessen if not wholly prevent smoke. With the chain grate type of mechanical stoker, instead of a higher rate of evaporation per pound of coal, a positive loss over hand-firing may result unless the stoker design prevents large excess quantities of air. If an analysis of the flue-gases shows 100 per cent. or more excess air, steps should be taken to prevent this air from entering the furnace, otherwise the economy will be greatly reduced.

Any type of mechanical stoker purporting to feed the fuel regularly into a properly designed furnace should furnish a solution to the problem of smokeless combustion, since, with uniform fuel supply, and the air under control, it ought to be possible to attain just that proportion between the two which is necessary for perfect combustion; and once having attained it, to *maintain* it. Practically this degree of perfection is not always realized, although mechanical stokers properly managed will often give results superior to ordinary hand-firing both in point of smokelessness and fuel economy, and permit use of lower grades of fuel than would be profitable with hand-firing.

WOOD.

The efficient burning of wood requires a large combustion chamber, and grates arranged to prevent admission of surplus air. The Stirling furnace perfectly meets these requirements, and is easily modified to suit any kind of wood fuel.

For the burning of shavings and sawdust, chutes are arranged in the boiler front, and feed the fuel directly upon the grates. Where sawmill refuse is conveyed to the boilers by carriers, a simple extension of the furnace provides a roof containing an opening through

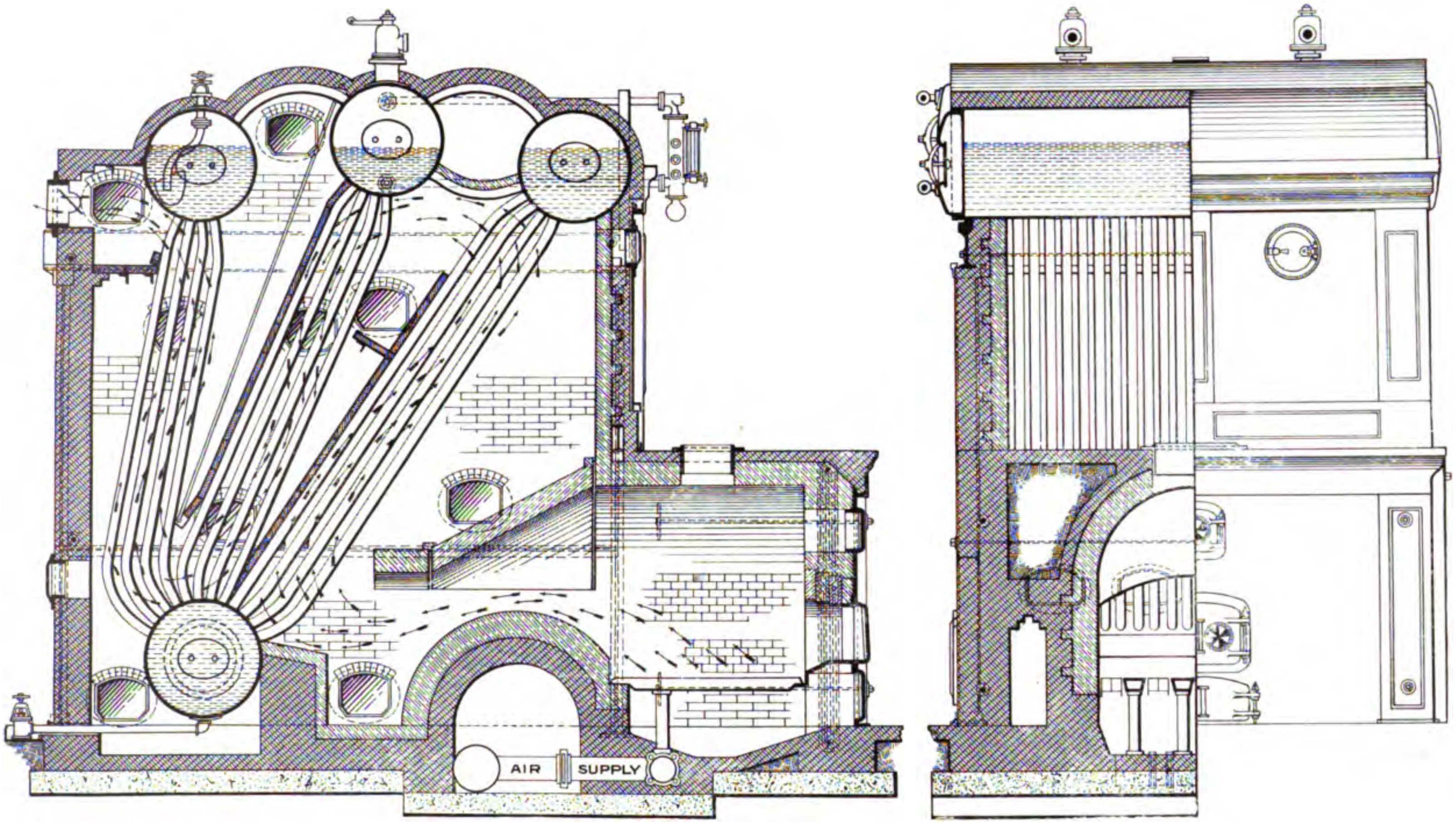


FIG. 30. SECTIONAL SIDE AND FRONT ELEVATION OF STIRLING BOILER AND BAGASSE FURNACE

which the refuse can be dropped automatically into the furnace.

For ordinary air-dried cord wood the grates are placed at firing floor level, their area is reduced to about two-fifths the amount required for coal, and the furnace walls, beginning under the arch, are battered to form a V, with the grates at the bottom. Cord wood to a depth of 30 to 36 inches can be carried on the grates, and the freshly fired wood crowds down that which has been partly burned, thus filling the large interstices at the bottom with burning coals, hence leakage of air past the fire is prevented. When other considerations prevent battering the furnace walls, the grates may be lowered as before, to secure the requisite thickness of fire, and the rear part of the grates may be blocked off, leaving in front the area that is desired.

For burning green cord wood and wet slabs, the conditions closely approximate those for burning green bagasse, as described in the following article, and a similar furnace may be used for both.

The Stirling Company has worked out many special arrangements of furnace for burning wood fuel in various degrees of dryness, and forms in which it is delivered as refuse from factories and mills, and is prepared to submit a design to fit any special conditions which may arise.

Tan Bark, or mixtures of tan bark, sawdust and slabs, are burned perfectly in the Stirling bagasse furnaces. In many cases such material containing as high as 55 per cent. of moisture is handled in this furnace with entire success, and the full rating of the boiler is developed notwithstanding the high content of moisture in the fuel.

BAGASSE.

Effect of Moisture—Though it has been shown in the chapter on Fuels that bagasse has practically a constant heat value per ton of the original cane, irrespective of the degree of juice extraction, it does not follow that bagasse of the low extractions will produce as much *useful* steam. To make this more clear, consider 2500 pounds of *dry* diffusion bagasse burned beneath a boiler, and assume that all of the heat of the bagasse ($2500 \times 8325 = 20{,}812{,}500$ B. T. U.) is generated and made available for evaporating water in the boiler; then with a boiler efficiency as low as 50% this heat would evaporate 10,773 pounds of water from and at 212°. But if to the 2500 pounds of dry bagasse, 7500 pounds of water be added, the mixture will not even burn unless dried or mixed with large quantities of dry fuel, notwithstanding the fact that the heat in the 2500 pounds of dry matter is sufficient to evaporate nearly three times the 7500 pounds of water present. Hence it is all important *how* the water is present. When mixed with the bagasse (and assuming that ignition is started) its evaporation results in the absorption of so much of the heat generated that the surrounding temperature is lowered to a point at which further combustion cannot take place, and the fire is extinguished. If this evaporation can be made to occur apart from the combustion of the bagasse, there will be sufficient heat generated to evaporate the water content and leave an excess for useful work.

Furnace Requirements—A high furnace temperature must be maintained, and this is best accomplished by making the furnace entirely of fire-brick and locating it away from the boiler heating surfaces, so that combustion may be complete before the boiler surfaces are reached. Consequently an extension furnace, in conjunction with the Stirling boiler, as shown in Fig. 30, proves eminently satisfactory, and is a combination widely used in the cane-sugar countries.

Stirling Green Bagasse Furnace—The Stirling furnace for green bagasse is a greatly improved form of the Burt patent. It is rectangular in shape, and is made in various widths and depths according to the capacity of the boiler and the quality of the bagasse. The roof is arched and the entire interior of the furnace is lined with fire-brick. The fresh fuel is admitted through an opening at the top, being conveyed by a carrier to be later described. See Fig. 31.

Grates—The grate surface is composed of Hollow Blast Grate Bars, alternating with plain herring-bone or straight ribbed bars. The hollow blast bar is a rectangular casting, provided with openings on its upper face which are covered with a sliding plate known as the "blast-valve," by means of which the air is discharged into the furnace in nearly horizontal jets, and so directed that those of

one bar cross those of the bars adjacent. Thus the air supply not only is under perfect control, but the manner of its admission insures an excellent distribution throughout the mass of fuel. The alternate arrangement of hollow and ordinary grate bars renders the furnace capable of burning coal or wood very advantageously, either with or without forced draft.

Air from the blower is led to a cast iron pipe in the ash-pit, and from this connections are made to the hollow bars from below; where there are several boilers the air supply of each is under separate control, permitting any boiler to be operated independently.

Advantages—The fire-brick walls and the arch become white hot, thus storing heat, which radiates upon and dries the fresh charge of bagasse. The moisture thus evaporated mingles with the highly heated gases from the bagasse already in the furnace, and the actual burning of the fresh charge does not begin until all of its moisture has passed off. The air supplied is *dry* and the volatile matter is burned at the high temperature necessary for proper combustion, the whole operation taking place before the boiler heating surface is reached

The superior points of this arrangement are: (1) The discharge of the air *into* the fuel insures the combustion of the sugar and molasses contained in it, and prevents formation of clinker. Where the air is admitted in any other manner, incomplete combustion occurs, and the sugar and silica form a hard clinker which is an endless source of trouble frequently requiring for its removal a complete shut down of the plant one or two hours daily. (2) The air supply can be so regulated that no excess over that actually required need be admitted. This greatly increases the efficiency of combustion.

Stoking Arrangements—An important feature where there are several furnaces is the bagasse conveyor, the automatic features of which contribute materially to the economy of the plant. See Fig. 31.

The conveyor is supported on a structural steel framework, and runs in a trough made of steel plate. The carrier is composed of endless chains fitted with bars which engage the bagasse and convey it from the mill to the boilers. In the bottom of the trough are adjustable openings through which any desired charge of bagasse is automatically dropped into hoppers set over each furnace. The hoppers are fitted with valves operated automatically by means of cams on a shaft along the boiler fronts. Periodically each valve opens, then closes when a charge of fuel has passed into the furnace, very little free air being admitted.

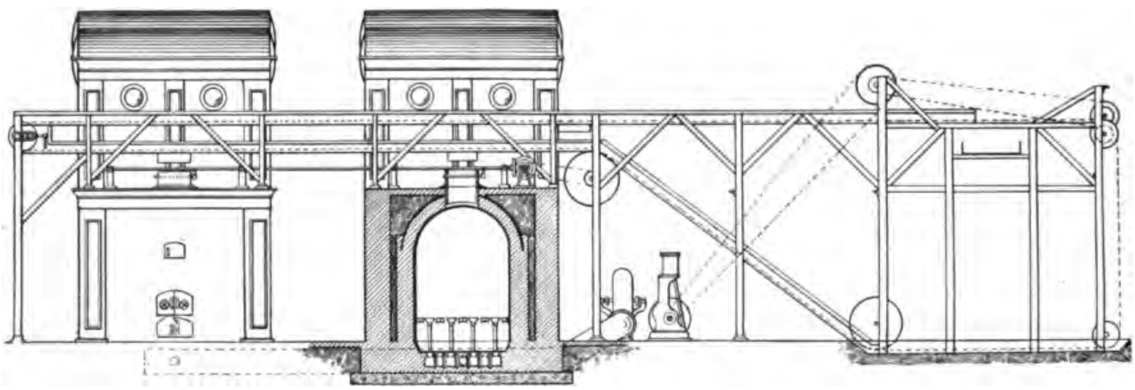


FIG. 31. FRONT ELEVATION OF STIRLING GREEN BAGASSE CONVEYOR AND AUTOMATIC FURNACE FEEDER

Excess Bagasse—Frequently more bagasse is discharged from the mill than is required at the time for steam-making; the conveyor provides for this by carrying the excess beyond the boilers, where it is stored until such times as it is needed. It is then conveyed back to the furnace by the same carrier. With this system of handling and burning green bagasse the fuel problem is greatly simplified, and a sugar-house can be operated with excellent economy. The apparatus is practically automatic, reducing

TEST OF STIRLING BOILERS BURNING GREEN BAGASSE

GENERAL DATA.

Date of test	December 26, 1896.
Duration of test	Six (6) hours.
Grate area, square feet	160
Heating surface, sq. ft.	5,750
Steam pressure (gauge)	98.1
Feed water	150.5° F.

FUEL.

Kind of fuel	Bagasse
Per cent. moisture	42.21
Total fuel consumed	67,343 lbs.
Total dry fuel consumed	37,577 "
Total refuse	566 "
Total combustible	37,011 "
Fuel burned per hour	11,224 "

WATER.

Total water apparently evaporated	153,178 lbs.
Total water actually evaporated	150,115 "
Equivalent actually evaporated from and at 212° F	165,682 "

ECONOMIC EVAPORATION.

Water evaporated per pound of bagasse from actual temperature and pressure	2.23 lbs.
Water evaporated per pound of combustible from actual temperature and pressure	4.05 "
Water evaporated per pound of bagasse from and at 212° F.	2.46 "
Water evaporated per pound of dry bagasse from and at 212° F.	4.41 "
Water evaporated per pound of combustible from and at 212° F.	4.50 "

RATE OF COMBUSTION.

Fuel actually burned per sq. ft. of grate surface per hour	70.2 "
Dry fuel actually burned per sq. ft. of grate surface per hour	39.1 "
Combustible burned per sq. ft. of grate surface per hour	38.5 "
Fuel combustible burned per sq. ft. of heating surface per hour	1.95 "
Dry fuel burned per sq. ft. of heating surface per hour	1.09 "
Combustible burned per sq. ft. of heating surface per hour	1.07 "

RATE OF EVAPORATION.

Water evaporated from and at 212° F. per sq. ft. of heating surface per hour	4.80 "
Water evaporated from and at 212° F. per sq. ft. of grate surface per hour	172.5 "
Water evaporated from 100° F. and 70 lbs. gauge pressure	144,121 "
Watcr evaporated from 100° F. and 70 lbs. gauge pressure per hour	24,020 "

COMMERCIAL HORSE-POWER.

H. P. rated at 30 pounds water per hour evaporation from 100° F. and 70 lbs gauge pressure	800
Builders rating in horse-power	500
Per cent. developed above rating	60

Mr. Pharr referring to this test, said: "The results will probably be considered extra good, but this test was made during an ordinary run and no precautions were taken to obtain favorable results."

labor costs to a minimum, and one man can operate six boilers, where ordinarily five or six men would be required.

Success of the Stirling System—The extensive application of the improved Burt furnace is evidenced by the fact that about four-fifths of the sugar plantations of Louisiana are equipped with it, and in almost every case in connection with Stirling boilers. The Stirling Company has installed many bagasse-burning outfits in the West Indies and Hawaiian Islands, and wherever sugar cane is grown Stirling boilers and Burt Bagasse Furnaces are the combination giving uniformly satisfactory results.

Test with Bagasse Fuel—The preceding test on Stirling boilers burning green bagasse, made by Mr. J. N. Pharr, shows an evaporation probably never before equaled with this fuel in this country. With the longer lived tropical bagasse even better results may be obtained. It is worthy of note that during the test the boilers were working at *sixty per cent. in excess of their rating.*

BURNING PETROLEUM

The requirements for the perfect combustion of petroleum are:—it must be thoroughly atomized and mixed with the requisite quantity of air; the mixture must be burned in a furnace constructed of refractory material, which will be durable under the high temperature developed, and radiate heat to assist in the combustion; and the combustion must be completed before the gases come into contact with the boiler tubes.

The first requirement is met by selection of a proper burner. The other requirements are so perfectly met in the Stirling furnace that the changes necessary from the design for coal are so few that in an hour after shutting off the oil burner the furnace may be made ready to fire with coal.

Fig. 32 shows the usual arrangement for oil burning. The rear half of the grate is covered with fire-brick laid close. In the front half of the furnace the bricks are laid with air spaces between them varying from $\frac{7}{8}$-in. wide directly under the burner tip, to $\frac{3}{8}$-in. wide at the line where the close brick begins. In the front half of the grate those portions not directly under the flame are covered with fire bricks laid about $\frac{1}{8}$-in. apart, hence the wider air spaces cover an area of V shape under the flame. A space $\frac{5}{8}$-in. wide is left at each side wall to admit air to cool the wall and promote combustion.

At the rear of the grate, or on the bridge wall, a checkerwork of fire-brick, from 14 to 18 inches high is usually introduced to break up the heat, and prevent it from striking directly upon the tubes. Owing to the recoil of the gases, and necessity for ample space in which they can expand, the fire arches terminate at a point about 24 inches from the nearest tube, measured at right angles to the tube. The spandrels of the arch should also be filled level so as to leave a throat of even width across the furnace. See Fig. 59, page 236.

In many cases the grates are omitted, and the ash-pit is filled with ashes or refuse fire-brick, up to a line connecting the top of the bridge wall and bottom of the ash-pit door, and this arrangement has given very satisfactory results.

When the grates are covered with fire-brick the burner is introduced through either the fire door, or a hole in the pier between doors; the burner tip is placed about 6 inches above the fire-brick over the grate, and projects the flame practically parallel with the grate. The air for combustion passes from the ash-pit up through the grates, absorbs heat in its passage through the layer of fire-brick, mixes with the atomized oil, and complete combustion ensues.

When the grates are omitted the burner is either placed as before, in which case it is directed downward slightly, or it is inserted on a level with top of ash-pit doors.

In either case the air supply is regulated by the ash-pit doors.

Direction of the Jet—If the proper quantity of air is supplied, the location or direction of the jet has no influence upon the combustion, but it has considerable influence on the efficient utilization of the heat. The experiments thus far made indicate that the nearly horizontal jet introduced from the boiler front, according to the methods above described, gives the best results. As soon as the heat is generated it is essential that it be absorbed by the boiler as rapidly as possible, without admixture of colder gases

which reduce the furnace temperature. In some cases burners have been inserted through an opening five to six feet above the floor line, and so pointed as to direct the flame downward; it then crosses the grates, reverses direction and travels up the front bank of meets the gases which have already been cooled by contact with lower part of the tubes, and the mixture of the two causes a reduction in temperature.

Heating the Air assists in the combustion, but usually the complication and expense

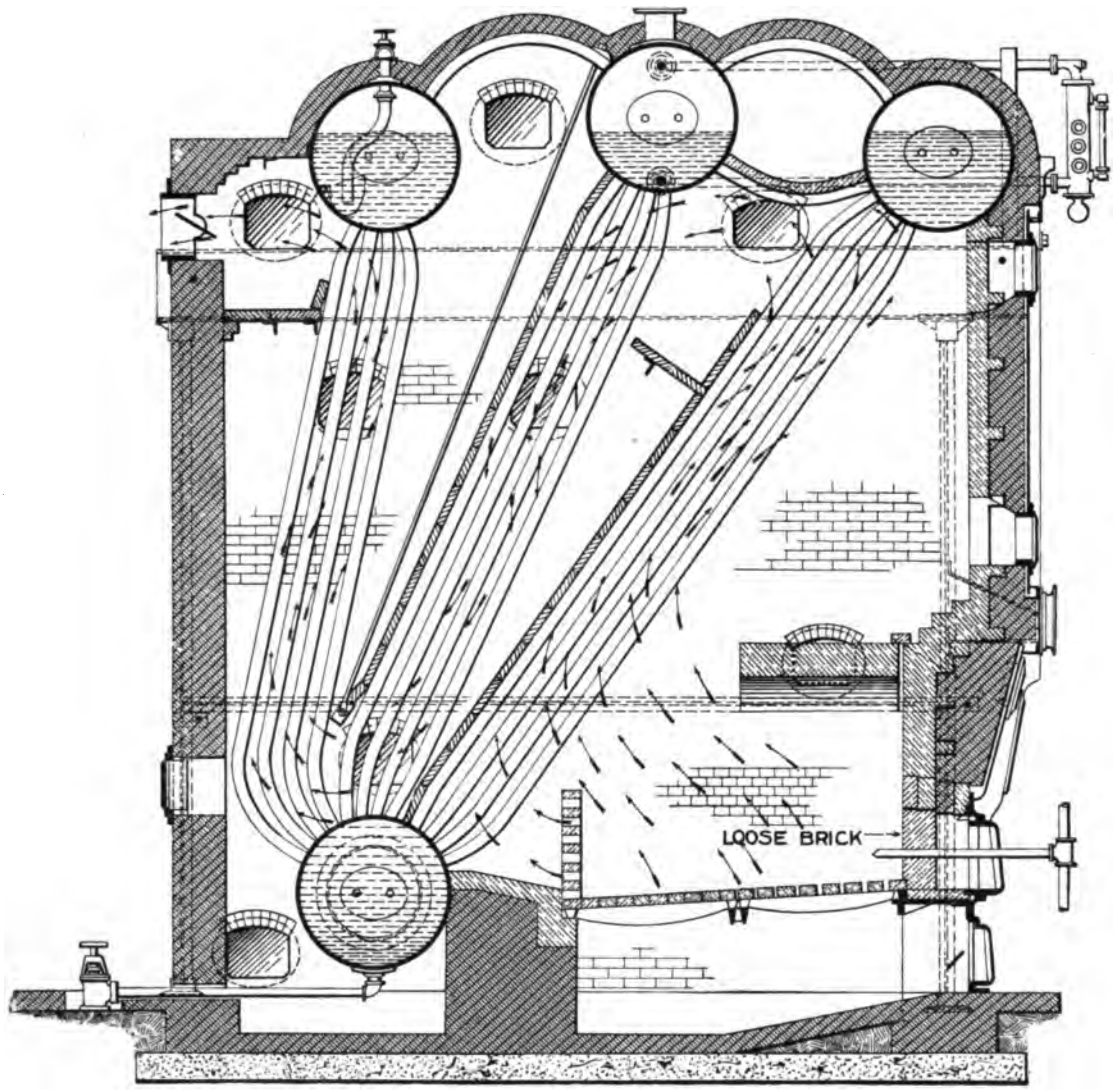


FIG. 32. SECTIONAL SIDE ELEVATION OF STIRLING BOILER FOR BURNING PETROLEUM OR NATURAL GAS

tubes. The fire arch is omitted. Experiments show that the efficiency is lowered by this arrangement, because it exposes a larger wall surface to the intense heat, thus causing greater loss by radiation; further, that part of the heat which rises from the front part of the flame directly to the top of the furnace, of doing it offset the advantage when steam spray burners are used. A simple and comparatively inexpensive method of heating the air is to provide the boiler with hollow walls; the air is drawn from the rear of the boiler, absorbs heat from the inner walls, and is admitted through ports in the side

walls, both above and below the grates. Installations of Stirling boilers thus arranged have proved entirely satisfactory.

Whatever arrangement be used the advantages of the Stirling furnace as described for coal apply with equal force to oil.

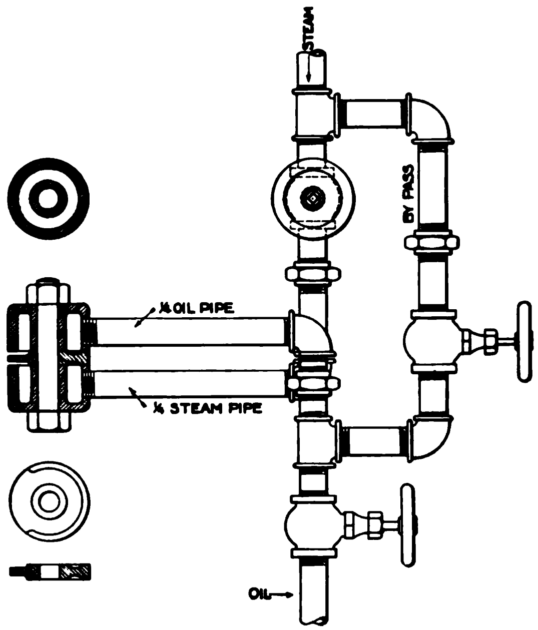


FIG 33 THE WARREN HYDROCARBON BURNER

Oil Burners—The function of the burner is to pulverize or atomize the oil to a condition approaching as far as possible that of gas, thus permitting the oil to be burned like a gas flame. Of the many hundred burners invented those in use may be reduced to two classes; (1) Spray burners, in which the spray is made by a jet of steam or compressed air; (2) Vapor burners, in which the oil is converted into vapor, then passed into the furnace.

While the vapor burner possesses merit, it has not come into general use. In spray burners the atomizing agent may be either steam, compressed air, or air and steam together. The steam spray burners are almost universally used; they are simple, require no blowers, compressors or other apparatus occupying space or demanding attention, and in the better types now obtainable at reasonable cost the steam used is so little as to be of less value than the expense of saving it.

Spray burners of the older types usually consist of two nozzles, one within the other; oil is fed through the inner and steam through the outer nozzle; the two currents meet and mingle and atomization is then effected. The disadvantage of the general arrangement is that the nozzles occasionally get clogged by dirt or formation of coke due to the heat, and the openings wear to a larger size than wanted. Accordingly the later types of burner dispense with the arrangement of one nozzle within another.

Steam spray burners are divisible into two classes: (1) Outside-mixers. (2) Inside-mixers. In the former the oil and steam meet outside the apparatus; the steam flows out through a flat slit or through a series of small holes in a horizontal row; the oil flows through similar slits or holes, and falls into the steam which seizes and atomizes it. Fig. 33 represents a burner of this type, invented by Mr. James W. Warren, of Los Angeles, Cal. Its construction is evident. The adjustment of the flame is easily made by filing the tip of the central washer, and wear is taken up by renewal of the washer.

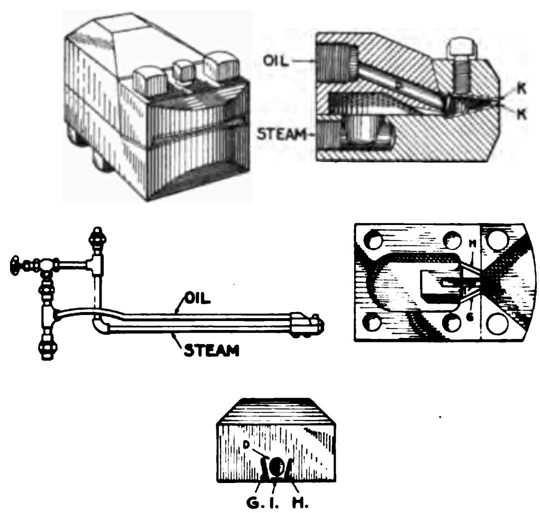


FIG. 34. HAMMEL OIL BURNER

In the inside-mixers the oil and steam mingle inside the apparatus and the mixture is atomized by passing through the nozzle Fig. 34 represents a burner of this type, invented by Mr. Chas. A. Hammel, of Los Angeles, Cal. The usual inner and outer nozzles are eliminated, and wear is provided

for by the removable plates *K-K*. The oil passing through the hole *D* is atomized by steam jets through the slots *G*, *H*, and *I*. In burners of this type the oil requires only sufficient heating to enable it to be pumped through the oil supply system, and oils or even tar, as low as 8° Beaumé can be successfully handled owing to the large oil channels.

In burners of the outside-mixer type the oil should be heated. This is usually done by passing the oil through an exhaust steam heater. The action of the burner is improved as the temperature of the oil is increased, up to about 210 F. If raised higher the water in the oil will vaporize and cause the flame to sputter. The steam supply is also frequently superheated by passing the steam pipe either into the furnace or the boiler breeching. When this is not done adequate provision should be made to drain out entrained water before the steam reaches the burner. A by-pass between steam and oil supply pipes should be provided to enable the oil ducts to be occasionally blown out with steam. See Figs. 33 and 34.

Regulation of Oil and Steam—When starting, the oil should first be turned on and ignited by some burning waste; then the steam should be turned on; the valves controlling the oil and steam should next be regulated so as to get the proper mixture. This regulation can best be accomplished by observing the top of the smoke stack, and the color of the fire. If the supply of steam is too great, steam will be seen surrounding the burning spray and issuing from the smoke stack; if the steam supply is deficient the atomization will not be completed; if the air supply is deficient the color of the flame will become red, and smoke will issue from the stack, indicating incomplete combustion. When the burner valves and the air supply are correctly adjusted the flame is a bright white in color and there is no smoke. Scintillating sparks indicate imperfect atomization.

Number of Burners Required—This varies with the type of burner. With either of the two burners already described, a furnace 8 feet wide can be served by one burner, a furnace 14 to 16 feet wide by three burners and intermediate sizes by two burners. The essential point is to distribute the heat evenly throughout the furnace, and evidently the more perfectly the burner forms a wide fan-tail flame, the fewer the number of burners needed.

Oil Pressure—This varies from a few pounds up to 60 pounds or over depending upon the type of burner. The oil system is usually fed by an ordinary duplex steam pump, with a relief valve between the suction and discharge sides. This is set at the desired pressure, and that pressure is always kept on the oil line to insure uniform supply through the burner. The oil passes from the pump through the heater then to the burner. The oil system should be provided with an air chamber to neutralize pulsations of the pump. In cold weather steam is circulated through pipes in the oil tanks, to keep the oil in condition to flow freely.

Per Cent. of Steam Used—In a series of tests made by the Bureau of Steam Engineering, U. S. Navy,* it was found that with a burner using air as an atomizing agent, the amount of steam required to compress the air varied from 1.06 to 7.45% of the total steam generated, the mean of eight tests being 3.18%. Four tests on steam spray burners varied from 3.98% to 5.77%, the average being 4.8%. Two tests on burners using steam and air together showed 8.54% and 6.09% respectively. In a series of most careful tests made for The Stirling Company on latest type of steam spray burner the results ran from 2.10% to 3.42% averaging 2.69% for four tests. It therefore does not seem that any saving of steam is to be made by employing air as the atomizing agent, and the use of steam obviates complication, and risk of interrupted service.

Boiler Efficiencies obtainable with Oil Fuel—Since oil can be burned with admission of but little more than the amount of air necessary to furnish the actual oxygen necessary for the combustion, and the furnace doors need never be opened while the boiler is under steam, and the boiler heating surface does not get quickly fouled by soot, it follows that with proper burners and careful attention higher boiler efficiencies may be expected with oil than with coal. It is highly important to keep down the content of water in the oil. The following tests on the Stirling boiler indicate the efficiencies obtainable with oil fuel. It should be noted that the per cent.

**Report of the Hohenstein Boiler and Liquid Fuel Boards.* U. S. Government Printing Office, 1902.

PART OF 3,000 H. P. OF STIRLING BOILERS BURNING OIL, THE LOS ANGELES GAS AND ELECTRIC COMPANY'S PLANTS, LOS ANGELES, CAL.

of water in the oil was large, hence with a smaller content of water even higher efficiencies would have been developed, illustrating the importance of sufficient tankage to allow the water to settle out.

Such high efficiencies cannot, however, be obtained with boilers that are not particularly adapted to use of oil fuel. This is well shown by the tests made by U. S. Bureau of Steam Engineering already referred to. There were four tests at rates of evaporation per square foot of heating surface equal to 3.91 5.18; 5.52; and 5.82 pounds of water from and at 212°. The corresponding boiler efficiencies were only 68.9; 71.5; 69.9, and 66.7%. The boiler was of the water-tube type containing 2130 square feet of heating surface, and operated at about 274 pounds pressure.

BURNING NATURAL GAS.

Practically the only difference between burning petroleum and natural gas is that the former, being liquid, must be atomized before it is mixed with the air requisite for combustion, while the latter, without any change of state, is ready to be mixed with the air and ignited. Consequently the burners will differ, but in other respects the form of furnace, length of fire-arches, location of the checkerwork wall in rear of the furnace, location and height of the burner above the fire-bricks covering the grates, and the air spaces between these bricks, will be the same for natural gas as for petroleum, hence the design of furnace shown in Fig. 32 will apply equally well to both.

TESTS OF A STIRLING BOILER OF 500 HORSE-POWER

AT PLANT OF THE LOS ANGELES GAS AND ELECTRIC COMPANY, LOS ANGELES, CAL.

Name of boiler		Stirling.	Stirling.
Heating surface	square feet,	5,020	5,020
Date of test, 1902		Nov. 13	Nov. 15
Duration of test	hours,	7½	5
Steam pressure, by gauge	lbs.	120	115
Temperature of feed water	Fahr.	143°	123°
Factor of evaporation		1.1157	1.1359
Pressure of oil	lbs. per square in.	24.7	31.4
Temperature of oil	Fahr.	198	193
Temperature of escaping gases	Fahr.	518	547½
Per cent. moisture in steam		0.54	0.54
Total water apparently evaporated	lbs.	117,960	97,872
Total water evaporated to dry steam	lbs.	117,323	97,343
Equivalent total water into dry steam, from and at 212°	lbs.	131,037	110,697
Kind of burner used		Warren	Warren
Kind of oil burned		Los Angeles	Los Angeles
Per cent. of water in the oil		9.87	9.16
Heat value of oil as fired, per lb.	B. T. U.	17,122	17,241
Heat value of oil, freed of all water	B. T. U.	18,997	18,979
Total oil as fired	lbs.	9,154	7,460
Horse-power developed		523.9	641.4
Horse-power, builders' rating		500	500
Per cent. developed above builders' rating		4.78	28.28
Water evaporated from and at 212° per lb. of oil as fired,	lbs.	14,314	14,839
Water evaporated from and at 212° per lb. of oil freed of water	lbs.	15.81	16,335
Efficiency of boiler		80.76	83.14
Average per cent. above rating		16.53	
Average efficiency of boiler		81.95	

NORTHERN TEXAS TRACTION CO., HANDLEY, TEXAS, OPERATING 1,200 H. P. OF STIRLING BOILERS BURNING OIL

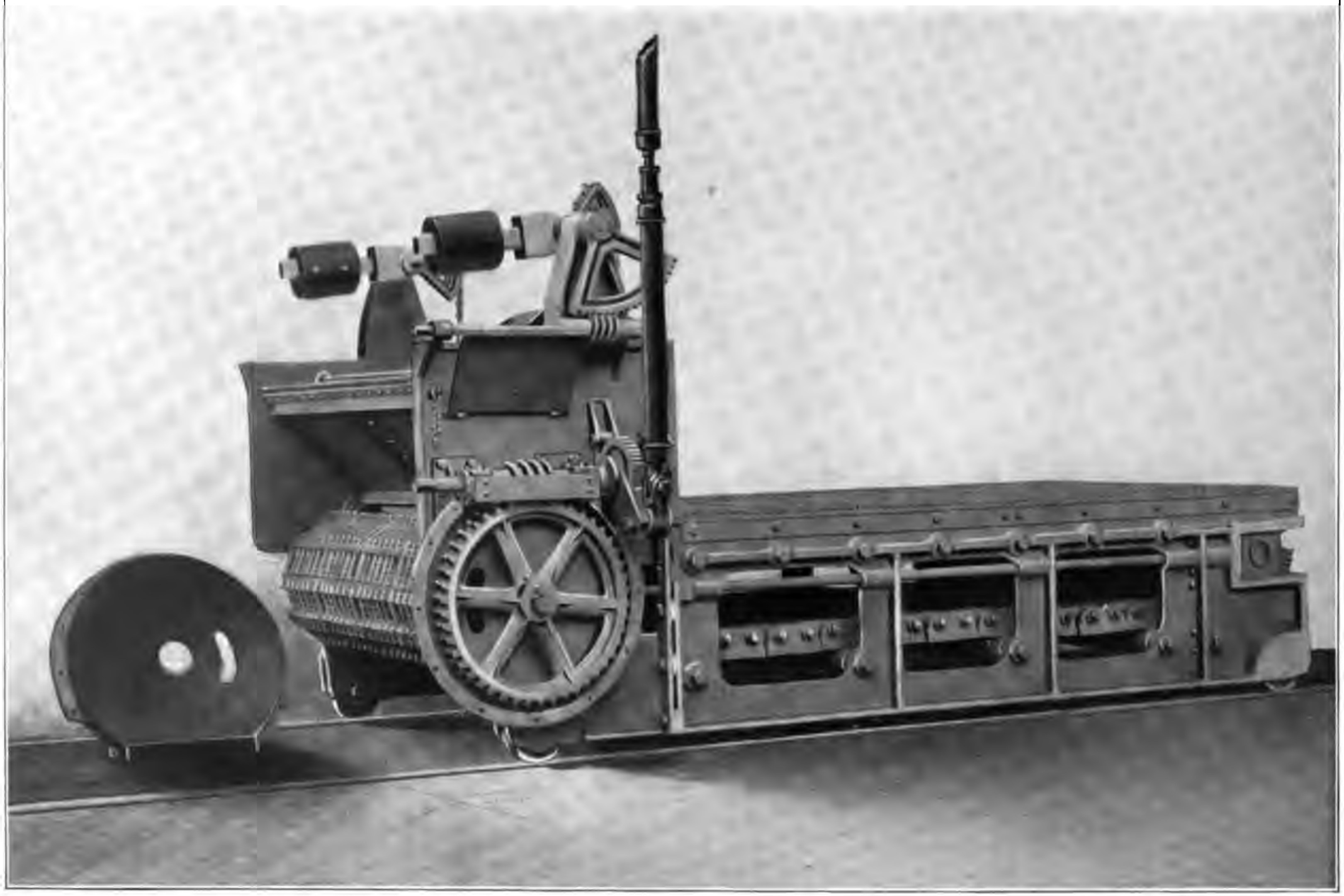
The most efficient gas burner will be that one which most intimately mingles the gas and air. A crude form of burner often used consists of a piece of one-half inch gas pipe placed inside of a piece of 2½-inch pipe which is bricked in the fire door opening. The suction created by the gas which blows out under pressure draws through the annular space between the two pipes a portion of the air needed for combustion, and the additional air required passes up through the slots between the bricks which cover the grates. The different types of burners on the market are all designed for the purpose of effecting a more intimate mixture of the gas and air than can be accomplished by the simple arrangement just described. The quantity of gas fed to the burner is regulated by an automatic reducing valve which is controlled by the steam pressure, and this is usually placed in the pipe which supplies gas to all or a number of the boilers, the meter being placed between reducing valve and burners.

The above test indicates the high efficiency obtainable from Stirling boilers fired with Natural Gas.

TEST OF A STIRLING BOILER BURNING NATURAL GAS

COLUMBUS, BUCKEYE LAKE, AND NEWARK TRACTION COMPANY, HEBRON, OHIO

Duration of test, hours		8
Pressures:	Steam gauge, pounds	140
	Draft in rear pass, inches of water	0.35
	Gas at meter, ounces	8
Temperatures:	Gas at meter	70° F
	Feed Water	116°
	Escaping flue-gases	417°
Kind of fuel		Natural Gas
Cubic feet of gas consumed		103,900
Cubic feet of gas consumed at 32° F. and 14.7 lbs. absolute		99,183
Total water used, pounds		77,826
Per cent. moisture in steam		0.65
Total water evaporated into dry steam, pounds		77,320
Factor of evaporation		1.1472
Water evaporated into dry steam from and at 212°, pounds		88,700
Water evaporated per cu. ft. of gas at standard condition, lbs		78
" " " " " " " " " " and from and at 212.° lbs		.895
Horse-power developed during the test		321
Horse-power, builders' rating		304
Per cent. horse-power developed above rating		5.6



THE STIRLING CHAIN GRATE STOKER

The Stirling Chain Grate Stoker

Chain grate stokers are extensively used for burning lignites, low grades of bituminous coal, and small sizes of anthracite. One of the most perfect stokers of this type is manufactured by The Stirling Company, and is illustrated in the photograph on the opposite page.

The stoker consists of a suitable framework, a travelling grate, fuel hopper, and the necessary driving mechanism.

The stoker is entirely self-contained, and no part of it is attached in anyway to the boiler framework or the brick setting; the entire machine rests upon four wheels which are supported by suitable rails which extend a sufficient distance into the fire-room to enable the stoker to be drawn completely from under the boiler.

The framework is made of cast iron and so designed that any part of it can be easily renewed. In the rear of the frame is a shaft upon which are placed idler pulleys, and a similar shaft in front is equipped with sprocket wheels. The grate is an endless chain composed of links of narrow width and relatively greater depth; this chain travels over the above mentioned pulleys and sprockets, and is driven by the sprocket shaft. The chain is of simple construction and any link can be replaced without disturbing other links. Between the front sprocket and the rear pulleys the chain is supported by cast iron rollers of narrow width, strung side by side on shafts extending across the framework. Any shaft with its rollers can be removed without disturbing any other part of the machine.

To provide for wear and expansion of the grate, the distance between centers of idler pulley shaft and sprocket shaft can be quickly adjusted even while the stoker is in operation. The feed gates are counterbalanced, and can be quickly adjusted to give any desirable thickness of fire. Inspection doors placed in the side of the furnace permit the condition of the fire to be noted, and a bar to be inserted when it is necessary to break up clinkers or remove obstructions to the free passage of the grates.

A common defect of chain grate stokers is the admission of excess air from the ash tunnel. In the Stirling chain grate this defect is overcome by the insertion of a diaphragm between the stoker and the ash-pit floor at a point just in front of the ash-pit tunnel. A suitably designed opening in the diaphragm permits the returning portion of the grate to pass through without admitting air, hence the air supply must come into the ash-pit from in front of the boiler, where it can be controlled in the usual manner.

The stoker may be driven from any convenient source of power, but the usual method is to operate it from an overhead shaft. A connecting rod driven from this shaft operates a crank on the side of the stoker framework; by means of a ratchet this crank moves a ratchet gear which drives the sprocket shaft through the medium of a worm wheel and a worm shaft attached to the ratchet gear. The ratchet may be adjusted to give the grates four different speeds, as occasion demands. An advantage of the Stirling chain grate is that all these working parts are housed in, thereby protecting them from dirt and grit. The connecting rod is so designed that in case of any obstruction tending to impede the motion of the grate, the driving mechanism stops, and breakage of parts is wholly obviated.

The size of air openings in the links, and other minor details, depend upon the character of fuel to be burned, and must be separately considered in each case. The Stirling Company is prepared to submit designs and estimates for chain grate stokers adapted to any conditions under which such stokers can be advantageously used.

ERECTING 300 H. P. OF STIRLING BOILERS, ILOILO ELECTRIC LIGHT & POWER CO., ILOILO, PHILIPPINE ISLANDS

Utilization of Waste Heat

A considerable saving of fuel and labor can be made by utilizing waste heat from blast furnaces, coke ovens, reverberatories for smelting ores, etc. While this fact has long been known, the installation of equipment for saving waste heat has not become so common as would naturally be expected, because of the lack of a boiler perfectly adapted to the peculiar nature of the work to be done. Boilers of the shell type do not absorb the heat readily, the available space is often too small to permit sufficient capacity of such boilers to be installed, and when the temperatures fluctuate considerably the shell type boiler causes trouble from unequal expansion. The requirements as to space can generally be met by installing water-tube boilers, but not all boilers of that type can comply with the other requirements. Whenever the boiler is out for cleaning, the heat which otherwise would be utilized by the boiler is usually wasted, hence it is essential that the boiler can be cleaned in the shortest possible time. The character of the gases also may vitally affect the boiler design. For example, the gases from reverberatory furnaces smelting copper matte contain a large content of sulphur, hence if the boiler develops a leak sulphuric acid is formed and the boiler plate is quickly destroyed. The sulphur fumes will penetrate each place where a leak occurs, therefore in those boilers using handhole caps the bearing surface and other parts affected by leakage from the caps will soon corrode, which explains why the cap type of boiler cannot be successfully used in connection with such furnaces.

These disadvantages are so completely obviated in the Stirling boiler that its merit as a waste heat boiler was quickly perceived, and its use for such work has rapidly increased. Not only has it met all requirements in a most satisfactory manner, but it is now operating with gratifying success under conditions of service which no other boiler has been able to meet. The perfect freedom from expansion obviates straining and leaks; the absence of caps and other complication eliminates the necessity of stoppage except for cleaning, and the time necessary for cleaning is less than required by other types as already shown.* The manhole plates are the only parts needing removal, and they are *completely outside the setting*, hence are not reached or affected by the gases. Large heating surface can be installed in the small space usually available, yet in no case need the general design of the boiler be changed, and should occassion demand it, the boiler can be removed and reset in the regular way. The form of furnace can be modified to conform to the requirements of the particular gas to be handled, and provision be made for hand firing when the supply of waste heat is cut off.

Each case requires careful study of all the conditions in order to determine the best method of utilizing the heat, and The Stirling Company will be pleased to confer with prospective customers, and to submit designs covering their requirements. The following descriptions will, however, indicate in a general way the amount of heat which may be saved, and the adaptation of the Stirling boiler to this class of service.

Coke Ovens—The best coking coal yields about 65 lbs. of coke per 100 lbs. of coal. Assuming the heat value of a pound of coke to be 13000 B. T. U., the coke produced by one pound of coal will represent 8450 B. T. U. The heat value of the coal would be about 13500 B. T. U., hence the heat loss during the coking process is 13500 − 8450 = 5050 B. T. U. Experience has shown that about one-half of this is lost by radiation from the oven and flue. Of the remainder about 70% can be utilized by a proper arrangement of flues and stack in connection with a boiler of good design and ample heating surface. Under these conditions the evaporation per pound of coal coked will be about

$$\frac{5050 \times 0.5 \times 0.70}{965.8} = 1.83 \text{ pounds.}$$

Assuming that one oven in 60 hours will coke 6 tons of coal, or 1,728,000 lbs. per year, and that if fired under the boiler direct one pound of the coal would evaporate 10 lbs. of water, then the annual saving in coal per

*See pages 20, 21 and 31.

oven under these conditions will be

$$\frac{1,728,000 \times 1.83}{10 \times 2000} = 158 \text{ tons.}$$

At fifty cents per ton at the mine this coal represents a saving of \$79.00 per year per oven, hence allowing 20% interest and depreciation, an investment of \$395.00 per oven for utilizing the waste heat would be warranted.

The arrangement of boilers, flues, etc., for a waste heat installation is very simple, but the details require close attention. The gas flues should be as short and direct as possible, and the stack must produce ample draft to draw the gases through the flues and the boiler. The stack height should be not less than 125 feet, or higher if the flues are long or crooked. The flue cross-section should contain .5 to .75 square feet per oven, which limits the number of ovens per flue to about 40. Each oven contributes enough heat to develop 10 to 12 boiler horse-power.

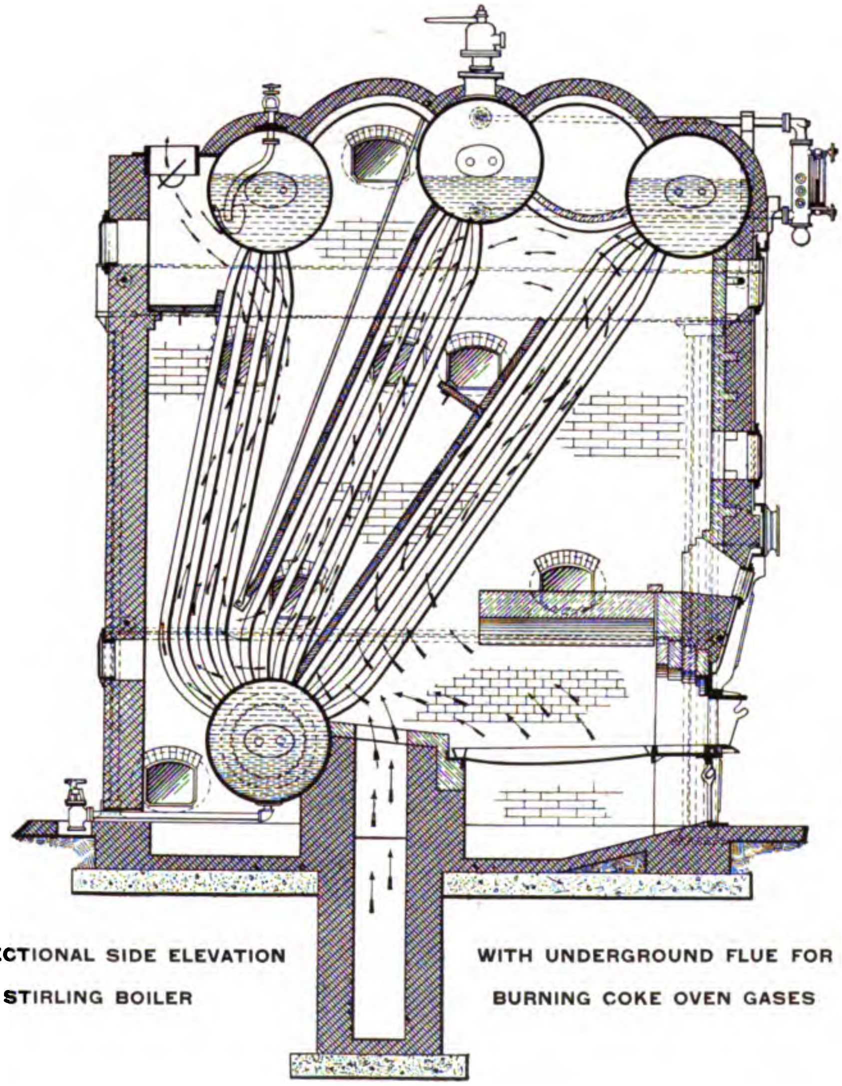


FIG. 35. SECTIONAL SIDE ELEVATION OF STIRLING BOILER WITH UNDERGROUND FLUE FOR BURNING COKE OVEN GASES

The foregoing statement applies more particularly to the bee-hive type of oven. In by-product ovens the heat loss is not so large, but even in these the waste heat can be utilized at a handsome profit.

Heating Surface Required—When the gases reach the boiler their temperature will not exceed 2000° and may be less. In a

boiler fired with coal the furnace temperature ranges from 2500° to possibly 3000°, hence the heating surface per boiler horse-power should be greater in the waste heat boiler than in the direct-fired boiler. From 12 to 15 square feet per horse-power will be needed for water-tube, and from 15 to 20 feet for return tubular, boilers. Since the volume of gas is large and its temperature comparatively low, a long pass through which the gases travel at considerable velocity and are well broken up by baffles, is of marked advantage in utilizing the heat. This is well shown by the comparative tests on Stirling and Lancashire boilers as later given.

Fig. 35 shows the Stirling boiler with underground flue, conveying coke oven gases to the furnace. The gases enter through an opening which extends along the whole length of the bridge wall. In front of the bridge wall is a grate for firing with coal when the gas supply is deficient, but it is better practise not to fire with coal when the boiler is utilizing waste heat. If the waste gases do not generate sufficient steam, an additional boiler should be installed and fired exclusively with coal, to attain the best results.

Tests—The following tests indicate the amount of heat that can be saved, and the advantage of the Stirling as a waste heat boiler. Owing to a defective damper in the gas flue leading to the Stirling boiler, the leakage into the by-pass flue was sufficient to produce a temperature of 1440° in that flue. If the heat thus lost could have been passed through the Stirling boiler, even better results would have been obtained. These tests corroborate the statements made that the heating surface for best efficiency with waste heat boilers should be greater than for coal-fired boilers. Under ordinary circumstances the wate -tube boiler will work most efficiently at a rate of evaporation of 3.45 lbs. of steam from and at 212° F. per square foot of heating surface. In these tests the evaporation on the Stirling was 4.01 lbs. per square foot; if this had been reduced to 3 lbs. it is evident that the temperature of the exit gases would have been reduced, thus increasing the efficiency.

Blast Furnace Gases..—Each ton of iron produced in the blast furnace requires from 1,800 to 2,200 lbs. of coke, and the weight of gases produced will be five to seven times the weight of coke used. From 25 to 30 per cent. by weight, of these gases will be carbon monoxide (CO). From Table 47 page 133, the calorific value of carbon monoxide at 32° F. and at atmospheric pressure, is 339 B. T. U. per cubic foot, and 4,350 B. T. U. per pound. By burning these gases under a boiler it is possible to utilize a large per cent. not only

TESTS OF ONE STIRLING, AND TWO 28 FOOT × 8 FOOT LANCASHIRE BOILERS BURNING COKE OVEN GASES

VICTORIA-GARESFIELD COLLIERY, ROWLAND'S GILL, NEWCASTLE-ON-TYNE, ENGLAND

Boilers	1 Stirling Class A, Standard.	2 Lancashire Type, each 28′ × 8′
Number of Beehive coke ovens	22	37
Boiler heating surface	1,611 sq. ft.	1,796 sq. ft.
Boiler heating surface per oven	73.4 " "	48.6 " "
Water evaporated per hour from and at 212° F.	6,465 lbs.	8,503 lbs.
Coal coked by above ovens per hour	3,800 "	6,391 "
Water evaporated from and at 212° F. per oven per hour	294 "	230 "
Water evaporated from and at 212° F. per lb. of coal coked	1.7 "	1.33 "
Water evaporated from and at 212° F. per sq. ft. of heating surface	4.01 "	4.79 "
Approximate temperature of gas at point of entry to boiler	1,720° F.	1,700° F.
Approximate temperature of gas leaving boiler	650° F.	750° F.
Normal evaporation of boiler if coal fired in the ordinary manner	6,445 lbs.	12,500 lbs.
Percentage evaporation secured to a normal evaporation of boiler if coal fired	100.3%	68%

of the heat produced by burning the carbon dioxide but also of the heat stored in the other gases at the high temperature at which they enter the boiler furnace.

The Stirling boiler has in a most satisfactory manner met every requirement for ber which enables the gases to be thoroughly mixed with air, and completely burned. The grates can be used to assist in making steam when there is a short supply of gas, without making it necessary to burn an excessive amount of coal for this purpose.

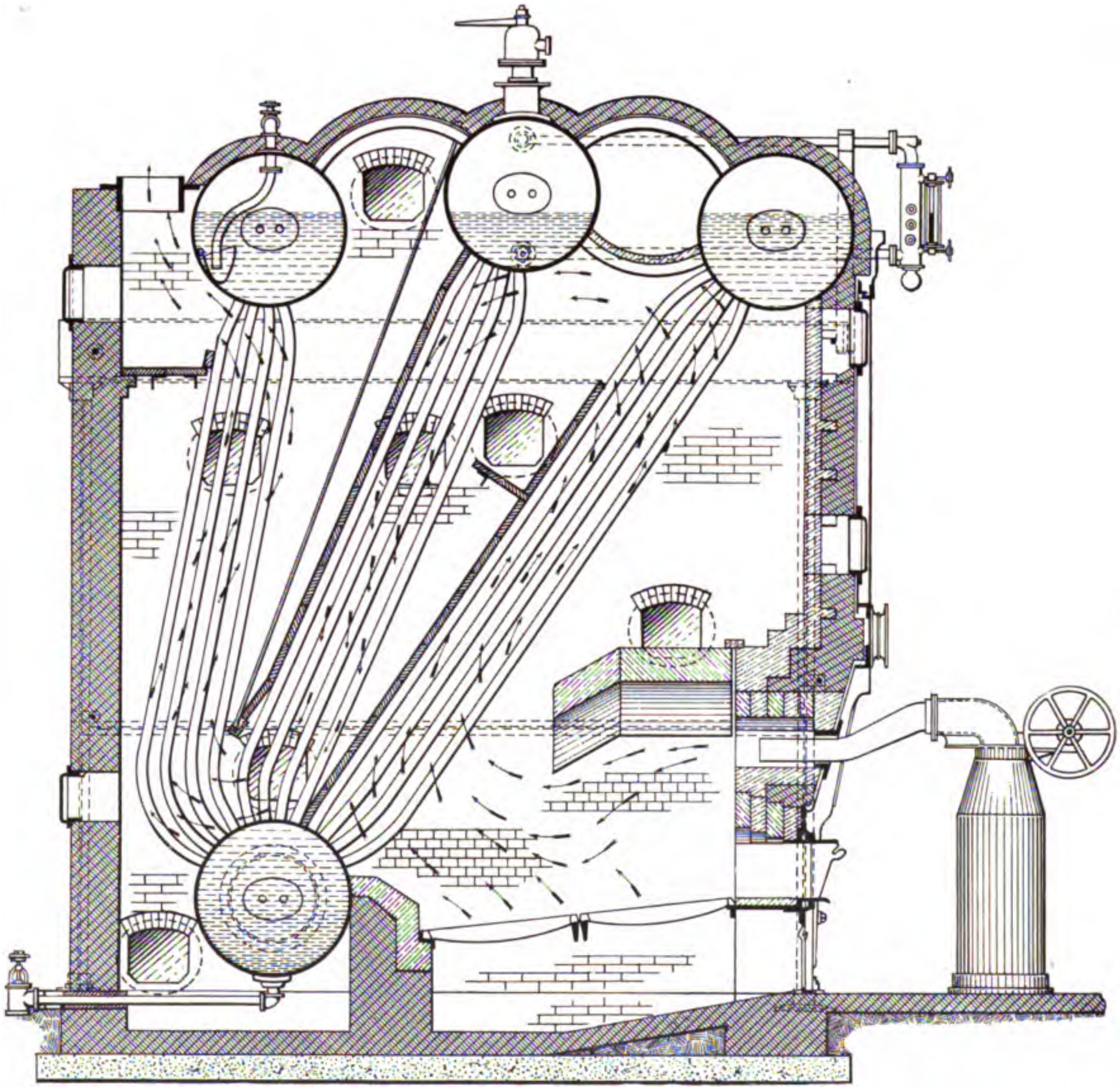


FIG. 36. SECTIONAL SIDE ELEVATION OF STIRLING BOILER FOR BURNING BLAST FURNACE GAS

burning blast furnace gases. Fig. 36 shows one of many designs adapted to such work. The boiler is coal-fired from the front in the usual manner, and the gases are brought into the setting at a point directly under the incandescent arch. The dropped arch at the rear of the furnace provides a large cham-

The setting is provided with cleaning doors so that any accumulation of dust can be readily removed without shutting down the boiler. By means of a cleaning door in the side of the setting in front of the middle bank of tubes, accumulations of lime, dust, etc., either on or between the tubes,

can be readily blown off. There is also an ample number of explosion doors, so that should there be an explosion of gas in the furnace, the setting would be immediately relieved of strains due to internal pressure.

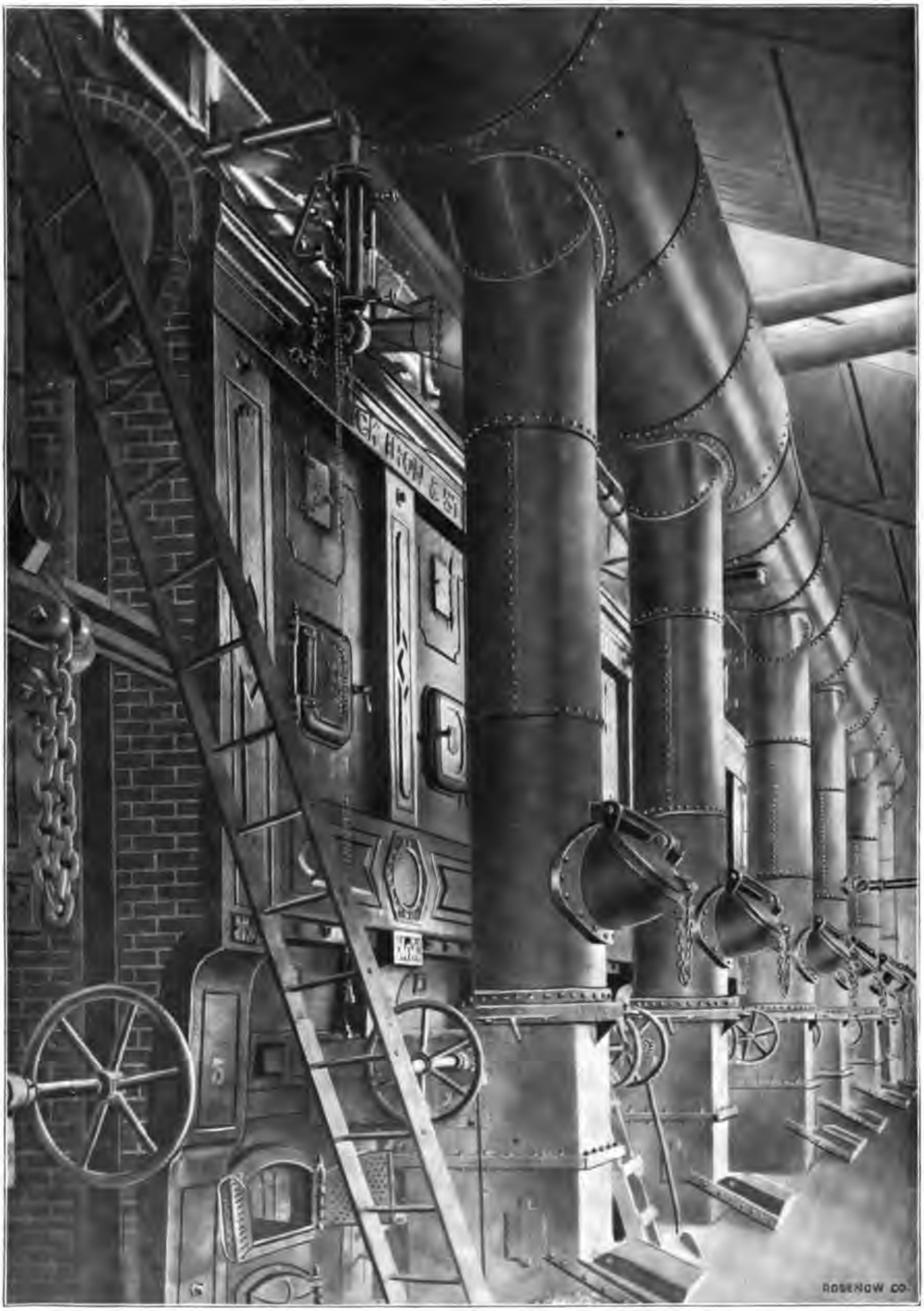
The following report of a test made on Stirling Boiler using blast furnace gases or fuel will prove of interest. Attention is directed to the very satisfactory results developed during the test.

TEST OF A STIRLING BOILER AT BRIER HILL FURNACE, YOUNGSTOWN, O.

USING BLAST FURNACE GAS AND COAL AS FUEL

Date of test			April 18, 1899
Duration of test, hours			7:00
Heating surface, square feet			2,788
Grate surface, square feet			52.64
Pressures:	Steam, pounds by gauge		80
	Barometer, inches		28.67
	Gas entering furnace, inches of water		0.917
	Draft at flue exit over damper, inches of water		0.45
	Draft at furnace, inches of water		0.25
Temperatures:	Gas at burners		322° F
	Escaping flue-gases		594°
	Feed water		55°
	Outside air		98°
Total gas used at burner temperature, cubic feet			1,397,907
" " " " 32° F " "			881,965
" coal " pounds			245
Heating value of gas, per pound at 32°		B. T. U.	1,082.63
" " " " cubic foot at 32°		B. T. U.	86.52
" " " coal used		B. T. U.	12,592
Total water used		pounds	50,380
Per cent. moisture in steam			0.6
Water evaporated into dry steam		pounds	50,078
" " " " "	from and at 212°	"	60,745
" " " " "	per 1,000 cu. ft. of gas at 32°	"	54.57
" " " " "	from and at 212° per 1,000 cu. ft. at 32°	"	66.19
" " " " "	per sq. ft. of heating surface per hour	"	2.56
" " " " "	from and at 212° per sq. ft. of heating surface per hour	"	3.11
Horse-power developed			251.5
Heat delivered to boiler per hour by combustion of	gas	B. T. U.	10,901,000
" " " " " " " " "	coal	"	440,720
" " " " " " " " "	gas and coal	"	11,341,720
" utilized in evaporation		"	8,382,948
Efficiency of boiler		per cent.	73.91
Efficiency of boiler not including hydrogen in heating value of fuel		" "	78.12

Analysis of fuel-gas	% BY VOL.	% BY WT.	Analysis of flue-gas.	% BY VOL.	% BY WT.
CO_2	13.50	20.00	CO_2	14.00	20.19
O	0.00	0.00	O	9.20	9.59
CO	25.20	23.62	CO	0.00	0.00
Hydrogen	1.43	0.097	Nitrogen and hydrocarbons	76.80	70.25
Nitrogen	59.87	56.25			
Specific gravity	1.032		Specific gravity	1.0603	



400 H. P. OF STIRLING BOILERS BURNING GASES FROM PUDDLING FURNACES, BLOCK-POLLACK IRON CO., CINCINNATI, O.

Furnaces for Smelting Copper—The gases from reverberatory furnaces smelting copper matte have an exit temperature which may reach or even exceed 2500°. The heat thus carried off represents a large per cent. of the calorific value of the fuel burned, hence the use of waste heat boilers at once suggests itself. The problem is, however, distinctly more difficult than when handling coke oven or blast furnace gases, because of the presence of a large content of sulphur in the gases. If these gases come into contact with water, sulphuric acid is formed, and the metal of the boiler is quickly destroyed by corrosion. The temperature also varies considerably during different stages of the smelting. In consequence, not only must the waste heat boiler be so designed as to secure perfect provision for expansion, thus obviating leaks, but it must also be free from handholes or other openings, which can be reached by the gases, and thereby be affected by the corrosion. The Stirling boiler meets these requirements perfectly. The curved tubes and suspended mud drum provide for free expansion and contraction; there are only four openings—one manhole in each drum,—and these are all *outside of the setting*, beyond the reach of the gases. In consequence, the Stirling boiler is perfectly adapted to use as a waste heat boiler in connection with furnaces smelting copper matte.

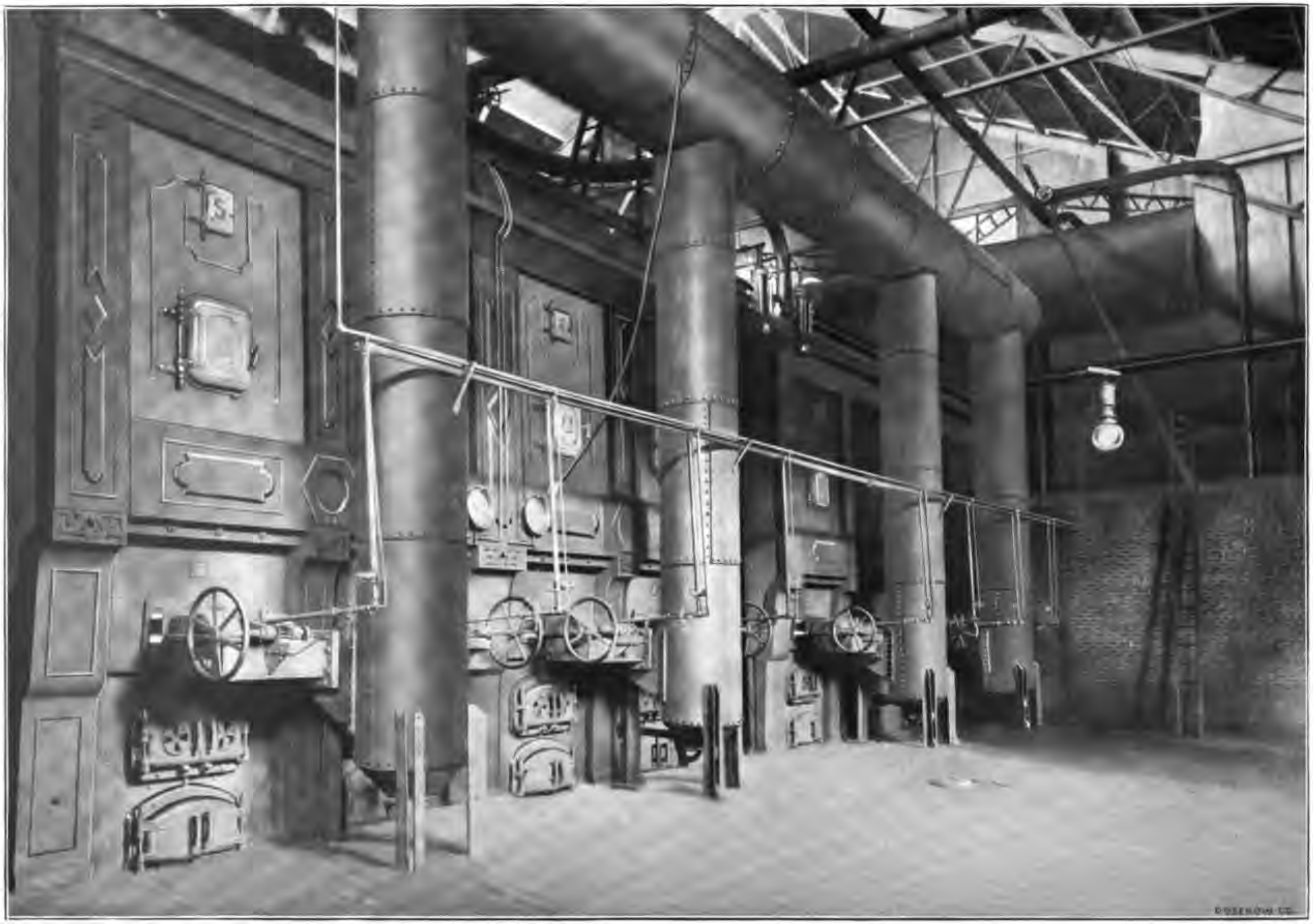
In Puddling and Heating Furnaces the metal to be heated absorbs only a small percentage of the calorific value of the fuel, and the remainder passes off with the furnace gases. A considerable portion of this heat can be saved by a properly designed waste heat boiler. The saving which can be effected is indicated by the following tests on Stirling boilers installed in connection with heating furnaces.

TABLE 50

SUMMARY OF TESTS OF STIRLING BOILERS INSTALLED IN CONNECTION WITH HEATING FURNACES

DATA OF TESTS.	AKRON IRON CO.	BLOCK-POLLACK IRON CO.
Heating surface, square feet	1438	1514
Grate surface, square feet	19.35	16.5
Ratio heating to grate surface	74	92.
Pounds of water evap. per hour	37.60	26.12
" " " " " " from and at 212° per square foot of heating surface	3.06	1.89
" " " " " " from and at 212° per lb. of coal	7.09	6.9
" " " " " " from and at 212° per lb. combustible	8.48	7.85

Waste Heat from Plain Cylinder Boilers.—Owing to the deficient heating surface of the plain cylinder boiler, the breeching temperature is excessively high when the boilers are forced. In consequence, this type of boiler is now fast going out of use, but where such boilers are still in good condition it has been found profitable to keep them in service, and utilize the heat they waste by passing the gases through a water-tube boiler, before turning them into the stack. The gases frequently leave the cylinder boilers at a temperature of 1500° to 1600°, and under such conditions the waste heat absorbed by the water-tube boiler will increase the capacity of the plant 75 to 100 per cent. without burning additional coal, or increasing the number of men employed.



PIONEER IRON CO., MARQUETTE, MICHIGAN, OPERATING 4,800 H. P. OF STIRLING BOILERS

Chimneys and Draft

The height and diameter of a chimney depend upon the kind and amount of the fuel to be burned, the design and the relative arrangement of the boilers and flues, and the altitude of the plant above sea level. Thus far no satisfactory formula involving all these factors has been produced, consequently empirical methods are used. In this chapter a method sufficiently comprehensive and accurate to cover all practical cases will be developed and illustrated.

Draft is the difference in pressure which causes gases to rise in a stack. If the air inside a stack be heated, each cubic foot of it will expand, hence its weight will be less than that of a cubic foot of colder air, therefore the unit pressure at the stack base due to the column of heated air will be less than that due to a column of cold air of equal height. This difference in pressure, like the difference in head of water, causes a flow of cold air into the base of the stack. But if in its passage to the bottom of the stack the cold air has to pass through a fire, it in turn becomes heated, hence it also will rise, and the action will be continuous.

The difference in pressure, or intensity of draft, is usually measured in inches of water.

Assume that the atmosphere has a temperature of 62° F. and the temperature of the gases in the chimney is 500° F. Neglecting for the present the increased density of the flue-gases as compared to air, the difference between the weight of the external air and internal flue-gases per cubic foot is .034 lbs., obtained as follows:

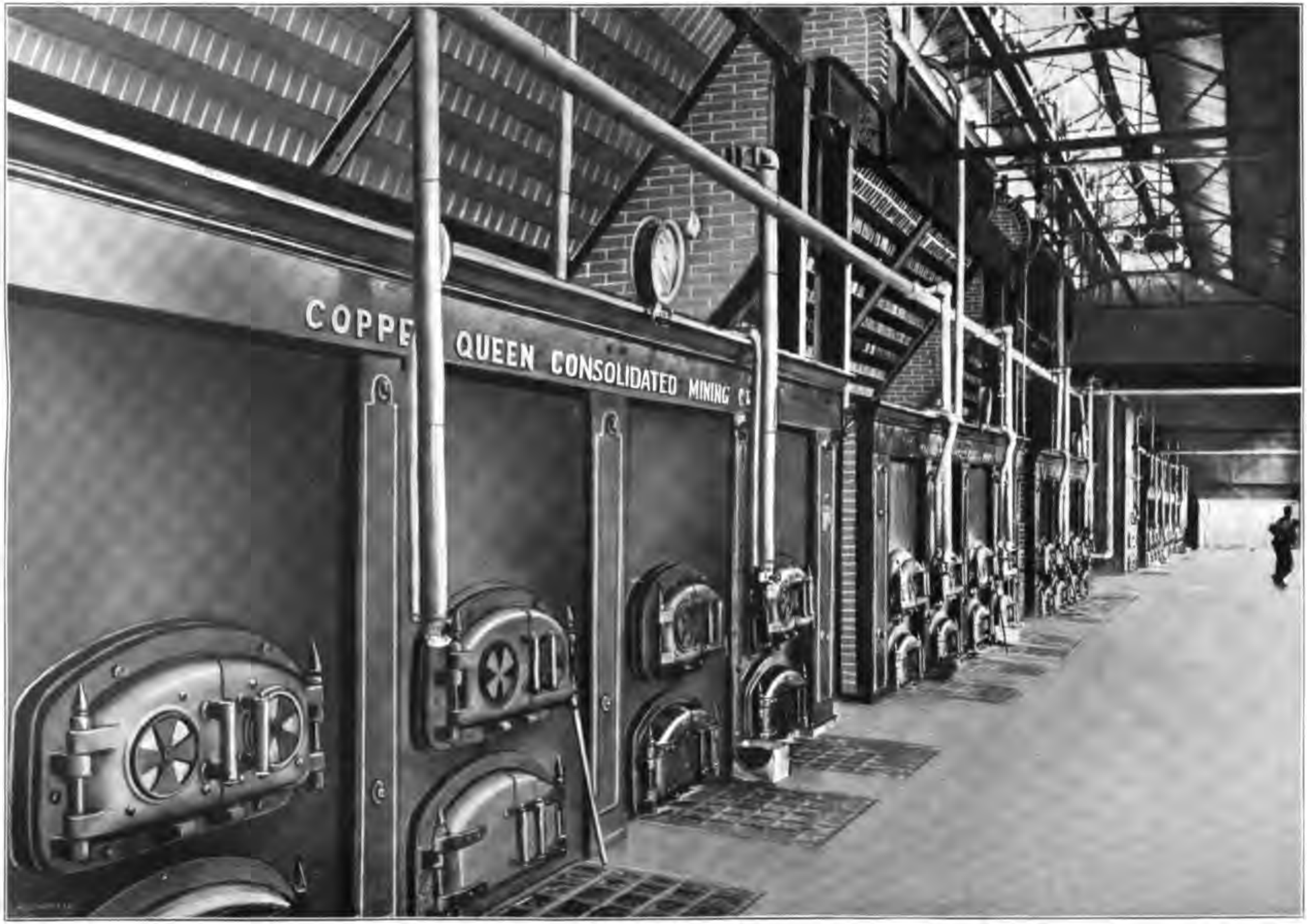
TABLE 51

THEORETICAL DRAFT PRESSURE IN INCHES OF WATER*
IN A CHIMNEY 100 FEET HIGH

(For other heights the draft varies directly as the height.)

TEMP. IN CHIMNEY FAHR.	TEMPERATURE OF EXTERNAL AIR. (BAROMETER 30 INCHES.)										
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
200°	.453	.419	.384	.353	.321	.292	.263	.234	.209	.182	.157
220	.488	.453	.419	.388	.355	.326	.298	.269	.244	.217	.192
240	.520	.488	.451	.421	.388	.359	.330	.301	.276	.250	.225
260	.555	.528	.484	.453	.420	.392	.363	.334	.309	.282	.257
280	.584	.549	.515	.482	.451	.422	.394	.365	.340	.313	.288
300	.611	.576	.541	.511	.478	.449	.420	.392	.367	.340	.315
320	.637	.603	.568	.538	.505	.476	.447	.419	.394	.367	.342
340	.662	.638	.593	.563	.530	.501	.472	.443	.419	.392	.367
360	.687	.653	.618	.588	.555	.526	.497	.468	.444	.417	.392
380	.710	.676	.641	.611	.578	.549	.520	.492	.467	.440	.415
400	.732	.697	.662	.632	.598	.570	.541	.513	.488	.461	.436
420	.753	.718	.684	.653	.620	.591	.563	.534	.509	.482	.457
440	.774	.739	.705	.674	.641	.612	.584	.555	.530	.503	.478
460	.793	.758	.724	.694	.660	.632	.603	.574	.549	.522	.497
480	.810	.776	.741	.710	.678	.649	.620	.591	.566	.540	.515
500	.829	.791	.760	.730	.697	.669	.639	.610	.586	.559	.534

**The available draft* will be the tabular values less the amount consumed by friction in the stack. In stacks whose diameter is determined by Formula 40 the *net* draft will be 80% of the tabular values. Hence to obtain from the table the height of stack necessary to produce a net draft of say 0.6 inches, the theoretical draft will be $0.6 \times 1.25 = 0.75$ inches, which can be got with a stack 100 ft. high with flue-gas temperature of 420° F., and air temperature of 0° F., or a stack 125 ft. high when the air temperature is 60° F.



3,000 H. P. OF STIRLING BOILERS, COPPER QUEEN CONSOLIDATED MINING COMPANY, DOUGLAS, ARIZONA

Weight of a cubic foot of air at 62° F. . . . . =.0761 lbs.
Weight of a cubic foot of air at 500° F. . . . . =.0414 "
Difference . . =.0347 "

Therefore, a chimney 100 feet high would have on every square foot of its base cross-section an upward pressure of .0347×100 = 3.47 lbs. As a cubic foot of water at 62° F. weighs 62.32 lbs., one inch of water will exert a pressure of $\frac{62.32}{12}$ = 5.193 lbs. per square foot, or $\frac{5.193}{144}$ = 0.03607 lbs. per square inch. The 100 feet stack will, therefore, show a draft of 3.47 ÷ 5.193 = 0.67 inch of water, nearly.

For the determination of the proportions of stacks and flues The Stirling Company's procedure depends upon the principle that if the diameter of the stack is sufficiently large for the volume of gases to be handled, the intensity of draft will depend upon the height; therefore,

Select a height of stack which will produce the draft required by the character and amount of fuel to be burned per square foot of grate surface, then,

Determine for this stack the diameter necessary to handle the gases without undue frictional losses.

The application of these rules follows.

Draft Formula—The force or intensity of draft is given by the formula:

$$D=0.52\,H\times P\left\{\frac{1}{T}-\frac{1}{T_1}\right\} \qquad [36]$$

In which,

D= draft produced, measured in inches of water.
H= height of top of stack above grate bars, in feet.
P = atmospheric pressure in lbs. per sq. in.,
T = atmospheric temperature, absolute.
T_1= absolute temperature of stack gases.

In this formula account is not taken of the density of the flue-gases, it being assumed to be practically equivalent to that of air. The error is safely negligible in practise.*

The force of draft at the sea level—which corresponds to a pressure of 14.7 lbs. per square inch—produced by a chimney 100 ft. high, when the temperature of the atmosphere is 60° F., and the flue-gas temperature is 500° F., is

$$D=0.52\times14.7\left\{\frac{1}{521}-\frac{1}{961}\right\}=.67$$

Under the same temperature conditions this chimney at a pressure of 10 lbs. per square inch—which corresponds to an altitude of about 10,000 feet above sea level—would produce a draft of only

$$D=0.52\times100\times10\left(\tfrac{1}{521}-\tfrac{1}{961}\right)=0.45 \text{ inch.}$$

For future use it is convenient to tabulate values of the product.

$$0.52\times1.47\left\{\frac{1}{T}-\frac{1}{T_1}\right\}=K$$

for a number of different values of T_1 and [36] becomes

$$D=K\,H \qquad [37]$$

For an atmospheric pressure and temperature, respectively, of 14.7 lbs. and 60° F., which represent average conditions, the results are as follows:

TEMPERATURE OF STACK GASES.	CONSTANT K
750	.0084
700	.0081
650	.0078
600	.0075
550	.0071
500	.0067
450	.0063
400	.0058
350	.0053

Draft Losses—The force of the draft as determined from the above formula can never be observed with the draft gauge or any recording device, but if the ash-pit doors are closed and the measurements are taken at the base of the stack, there will be but little difference between the actual and the theoretical draft. The difference existing at other times represents the pressure required to force the gases through the stack against the friction of the sides and against their own inertia, and increases with the velocity of the gases. When the ash-pit doors are closed, the volume of gases passing is a minimum, hence the maximum force of draft is shown on the gauge.

As the measurements are taken farther along the path of the gases through the boiler,

*Some draft formulas are based upon the assumption that twenty-four pounds of air are used per pound of coal, hence the air will weigh 96% of the chimney gas. Table 51 gives the draft pressures in inches of water worked out on this hypothesis. Owing to the variation in the air supply the draft either from the table or from Formula [36] will be accurate enough for all practical purposes.

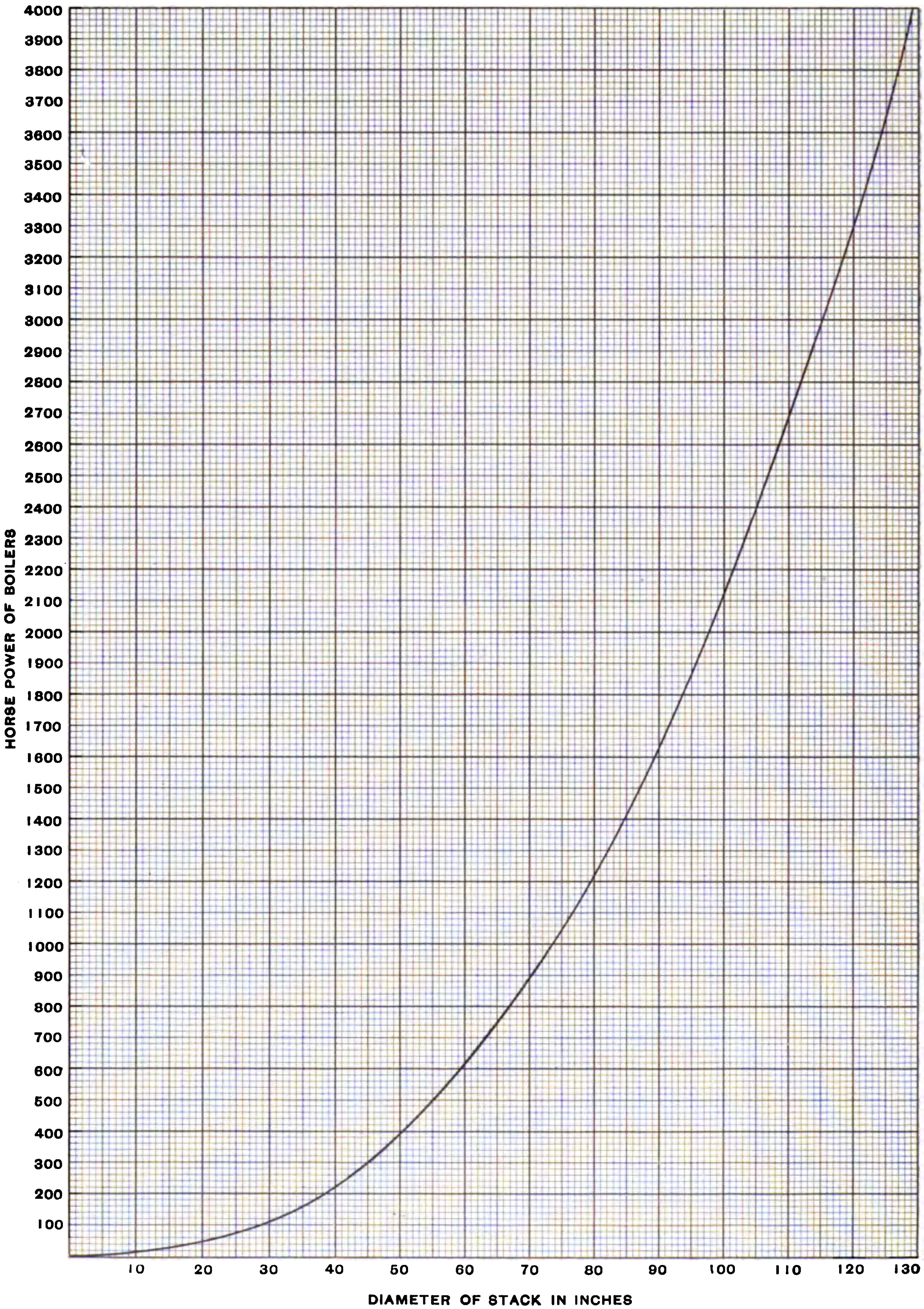


FIG. 37. CURVE SHOWING DIAMETER OF CHIMNEY STACKS AT SEA LEVEL

COMPUTED FROM FORMULA NO. 40. FOR BRICK OR BRICK-LINED STACKS, INCREASE THE DIAMETER 6 PER CENT. ONE-FIFTH OF THE THEORETICAL DRAFT IS LOST IN THE STACKS

from the stack toward the grate, the readings grow gradually less, until in the ash-pit hardly a perceptible rise takes place in the water of the gauge. The breeching, the boiler damper, baffles and tubes, and the coal in the grate, all retard the passage of the gases, and in each case the draft from the chimney is required to overcome their resistance. The draft at the rear of the setting, where the connection is made to the flue or stack, might be 0.5 inch, while in the furnace over the fire it might not be more than 0.15 inch, the difference, 0.35 inch, being the draft required to force the gases between the tubes and around the baffling.

An important factor in chimney design is the pressure required to force the air through the bed of coals. In many instances this will be a large percentage of the total draft. Its measure is found directly, in the case of natural draft, by noting the draft in the furnace, for it is evident with ash-pit doors of ample size the pressure under the grates will not differ sensibly from the atmospheric pressure.

Loss In Stack—The difference between the theoretical draft as determined by formula [37] and the amount lost by friction, etc., in the stack proper, is the *available* draft, or that which the draft gauge indicates when connected to the base of the stack. The sum of the draft lost in the flue, boiler, and furnace, must be exactly equal to the available draft, and as these quantities can be determined from records of experiments, the proportioning of a stack resolves itself into finding a stack which will produce a given available draft.

The loss in the stack and flue by friction and inertia can be calculated from the following formula:

$$\triangle D = \frac{fW^2CH}{A^3} \qquad [38]$$

where $\triangle D$ = draft lost in inches of water.

W = weight, in pounds, of gases passing per second.

C = circumference of a stack or flue in feet.

A = area of passage in square feet.

H = height of stack in feet; or when used for flues, length of flue.

f = a constant with the following values, for sea level:

.0015 for steel stack, temperature of gases 600° F

.0011 for steel stack, temperature of gases 350° F.

.0020 for brick or brick-lined stack, temperature of gases 600° F.

.0015 for brick or brick-lined stack, temperature of gases 350° F.

The available draft is equal to the difference between the theoretical draft from Formula [37], and the loss from Formula [38], hence

$$d' = \text{available draft} = KH - \frac{fW^2CH}{A^3} \qquad [39]$$

Height and Diameter of Stack—It follows from this formula that a stack of a certain diameter, by increasing its height, can be made to produce the same available draft as one of a larger diameter, the additional height being required to overcome the greater friction loss. Consequently, among the various stacks which could meet the requirements there must be one which can be constructed cheaper than the others. By deducing an equation connecting the cost of stacks with their height and diameter, and using it in connection with the formula for available draft, it has been found that the minimum-cost stack has a diameter depending solely upon the horse-power of the boilers it serves, and a height proportional to the available draft required.

Assuming 120 lbs. of flue-gas per hour for each boiler horse-power, which provides for allowable overload and use of poor coal, the method above stated gives:

For an unlined steel stack,

$$\text{Dia. in inches} = 4.68\,(H.P.)^{\frac{2}{5}} \qquad [40]$$

For stacks lined with masonry,

$$\text{Dia. in inches} = 4.92\,(H.P.)^{\frac{2}{5}} \qquad [41]$$

In both of these formulas $H.P.$ = rated horse-power of boilers.

From this formula the curve in Fig. 37 has been calculated, and from it the stack diameter for any boiler horse-power can be taken.

Stacks with diameters determined as above have an available draft which bears a constant ratio to the theoretical draft, and, allowing for the cooling of the gases in their passage up through the shaft, this ratio is .80. Using this correction in Formula [37], and transposing, the height of the chimney becomes

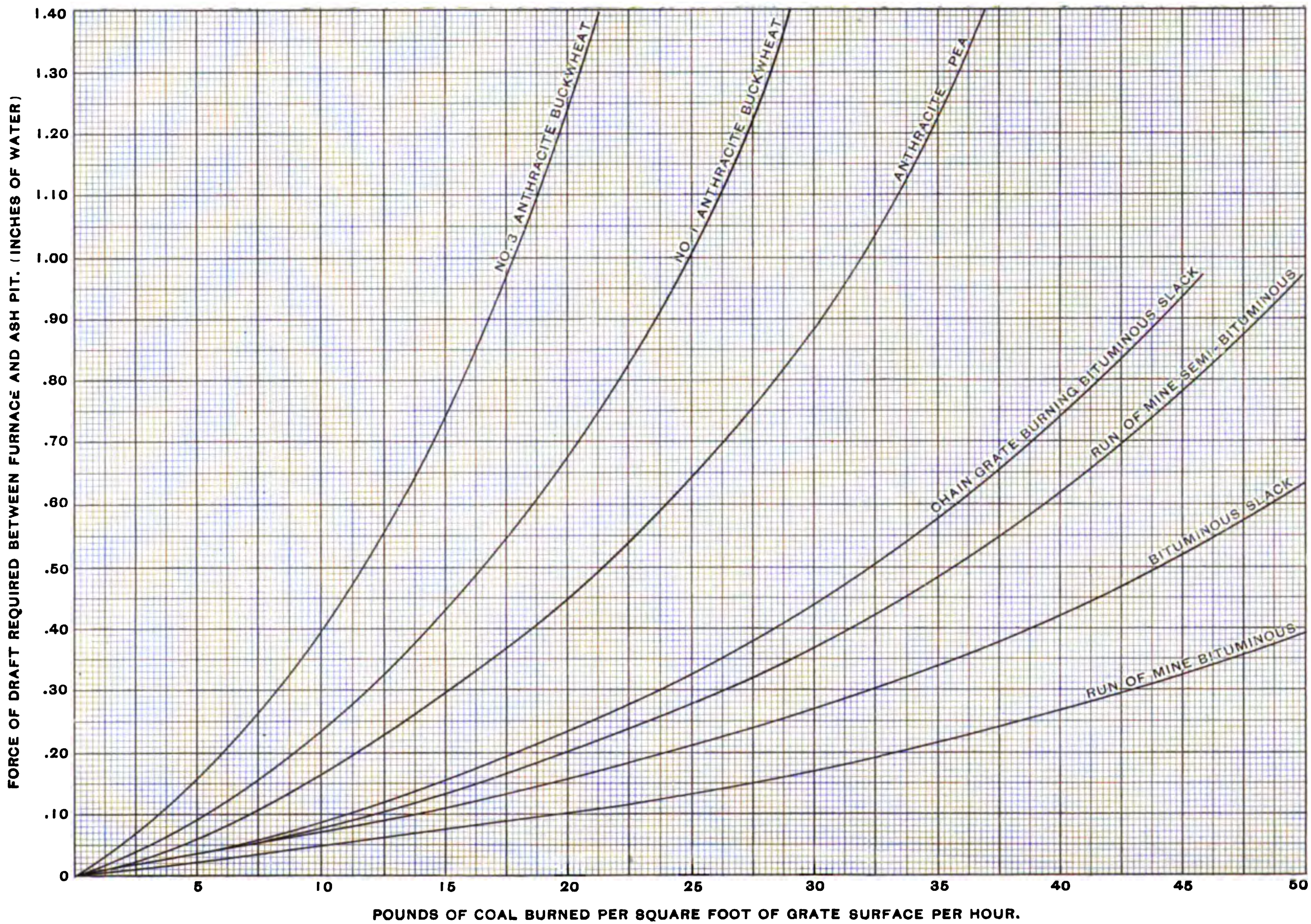


FIG. 38. CURVES SHOWING DRAFT REQUIRED BETWEEN FURNACE AND ASH-PIT AT DIFFERENT COMBUSTION RATES FOR VARIOUS KINDS OF COAL

$$H = \frac{d'}{.8\,K} \qquad [42]$$

H = height of stack in feet, measured from the point where the flue enters,
d' = available draft required,
K = constant as in formula [37].

Losses in Flues—The loss of draft suction in passing through a straight flue can be calculated approximately from Formula [38], which was given for the loss in a stack. It must be borne in mind that C in this formula is the actual perimeter of the flue, and is least compared to the area when the section is a circle, is greater for a square, and still larger for a rectangle. The retarding effect of the square flue is 12% greater than of a circular one of the same area. The greater resistance of the more or less uneven brick flue is provided for in the values given to the constants. Both steel and brick flues should be short, and as near to a circular or square section as is possible. Abrupt turns are to be avoided, but long, easy sweeps take up valuable space, and it is often desirable to add to the height of a stack, rather than take up additional room below. Short right-angled turns reduce the draft by an amount which can be roughly approximated as equal to 0.05 inch for each turn. The turns which the gases make in leaving the damper box of a boiler and entering a horizontal flue, must always be considered.

The sectional area of the passage leading from the boilers to the stack is determined largely by considerations of cost, and the subject resolves itself into whether it is cheaper to add to the height of the stack or to increase the flue area. The general practise is to make the area of the flue the same as, or slightly larger than, that of the stack; its area should preferably be at least 20% greater. It is unnecessary to maintain the same size of flue the entire distance behind a row of boilers, and the area may be reduced as connections with the various boilers are passed.

With circular steel flues of the same size as the stack or reduced proportionately to the volume of the gases, a convenient rule is to allow 0.1 inch draft loss per each hundred feet length of flue, and 0.05 inch for each right-angle turn. For square or rectangular brick flues, these values should be doubled.

Loss in Boiler—In calculating the available draft of a chimney, 120 lbs. per hour has been used as the weight of the gases per boiler horse-power. This covers an overload of the boilers to an extent of 50%, which provides for all practical requirements. Stirling boilers require comparatively little draft in the boiler proper, 0.2 inch being all that is lost when working at rated capacity. At 50% overload, 0.4 inch should be allowed, and this figure is the one to be used when summing up the available draft the stack must furnish.

Loss in Furnace—The draft loss in the furnace varies between wide limits. The air necessary for combustion must come through the interstices of the coal on the grate, and when these are large, as with a broken lump coal, but little pressure is required to force air through; but if they are small, as with slack or anthracite culm, a much greater pressure is required. If the draft is insufficient the coal will accumulate on the grate and a dead, smoky fire will result, causing imperfect combustion; if the draft is too great the coal is rapidly consumed, leaving a thin fire and portions of the grate bars uncovered.

Draft Required for Different Fuels—For every kind of fuel and rate of combustion there is a certain draft with which the best general results are obtained. It is comparatively small with the free-burning bituminous coal, and increases in amount as the percentage of volatile matter diminishes and the fixed carbon increases, being highest for the anthracites. Other things, such as the percentage of ash and the air spaces in the grates, etc., exert an influence, but like other factors, their effect can be found only by experiment.

The curves in Fig. 38 give the furnace draft necessary to burn various kinds of coal at the combustion rates indicated as abscissas. These have been plotted from the records of numerous tests in the files of The Stirling Company, and they allow a safe margin for economically burning coals of the kind noted. One curve is given for the draft required with Stirling chain grates burning bituminous slack. The greater

draft than that required for hand firing is due to the fact that in the chain grates the fire is not broken up and cleaned. As the amount of fixed carbon in coal increases, the differences in the draft required by a chain grate and for hand firing grow less, and for culm they are about the same; in both, however, the draft is so great as to necessitate very high stacks or forced draft.

Rate of Combustion—The amount of coal which can be burned per hour per square foot of grate is controlled by the character of the coal and the ratio of grate surface to boiler heating surface. When this ratio is properly proportioned the efficiency of boiler and furnace will be practically the same for different rates of combustion (unless they are either unduly large or small), provided the draft is adjusted to suit the particular rate of combustion desired. Hence the area of the grate can be fixed and the stack be designed to suit, or the stack may be decided upon, and the grate area be adjusted to burn the necessary quantity of coal at a rate per square foot of grate corresponding to the draft the stack can provide.

Solution of a Problem—The stack diameter can be determined from the curve, Fig. 37. The height is determined by adding the draft required in furnace, boiler, and flue, and computing from formula [37] the height necessary to give this draft. Example: proportion a stack for 1000 H. P. of boilers, using chain grates, burning fuel that will evaporate 8 pounds of water from and at 212° per pound of coal; ratio of heating surface to grate surface being 40 to 1; the flue being 100 feet long, with two right-angled turns; the chimney to be able to handle boiler overloads of 50%.

The atmospheric temperature may be assumed as 60° F., and the flue-gas temperature at the stated boiler overload as 550° F.

The combustion rate at boiler rating is

$$\frac{40\times 3\frac{1}{2}}{8}=17.5 \text{ pounds.}$$

For 50% above rating, the combustion rate will be about 60% more than this, or $1.60\times 17.5 = 28$ lbs. of coal per sq. ft. of grate surface per hour. The furnace draft required for this combustion rate, from the curve, Fig. 38, is 0.4 inch. The loss in the boiler also will be 0.4 inch, the loss in the flue 0.1 inch, and in the turns $2\times 0.05=0.1$ inch. The available draft required at the chimney where the flue enters is therefore:

Boiler	0.4 inch
Furnace	0.4 "
Flue	0.1 "
Turns	0.1 "
Total	1.0 inch

Since the available draft is 80% of the theoretical, the theoretical draft due to the height required is $1.00\div .8=1.25$ inch.

The chimney constant for temperatures of 60° and 550° F. is .0071, formula [37], hence the height of the stack above the point where the flue enters is, by the same formula

$$\frac{1.25}{.0071}=175 \text{ ft.}$$

Its diameter, from the curve in Fig. 37, is 75 inches if unlined, and 80 inches inside if lined with masonry. The greatest diameter of the breeching, if circular, for 20% greater area than the stack, would be 82 inches, and would taper down to about 42 inches where the last boiler connects, if four units were used.

Correction for Altitude—From formula [36] it follows that the draft is proportional to the atmospheric pressure, hence for a stack of given height the draft will decrease when the altitude is increased, consequently to secure at high altitudes the draft necessary for the rates of combustion in Fig. 38 the dimensions of a stack as determined for sea level must be altered.

Let p be the atmospheric pressure at sea level, and p_1, the pressure at any other altitude; H the height of a chimney which at sea level produces a given draft, and H_1, the height of a stack which will give at the assumed altitude the same draft that H gives at sea level. Put $\frac{p}{p_1}=r$, then from formula [36], $H_1=rH$. It therefore remains only to determine the increased diameter needed. Formula [18], page 87, shows that the weight of gas per minute flowing through a pipe is

$$w=\text{a constant}\times\left\{\frac{PDd_1^5}{\left\{1+\frac{3.6}{d_1}\right\}L}\right\}^{\frac{1}{2}}$$

Let d=diameter of the stack H determined for sea level; d_1 the diameter of H_1 determined for the altitude; D the density of gases at sea level, and D_1 the density at the given altitude. In the formula P will evidently be the quantity $\triangle D$, in formula [38] page 173, which will be the same in both stacks H and H_1; regardless of the altitudes the same weight of oxygen will be needed to burn a pound of a given coal, hence the diameter of stack H_1 must be such as to pass the *same weight* of gas at the altitude that stack H passes at sea level. To apply the formula, H and H_1, will be the lengths denoted by L; also, as previously shown, $H_1 = rH$ and $D = rD_1$, hence since both stacks deliver the same quantity of gas, it follows, neglecting the small term $\left\{1+\frac{3.6}{d_1}\right\}$, that

$$\frac{PDd^5}{H} = \frac{PD_1d_1^5}{H_1} \text{ or } \frac{D_1rd^5}{H} = \frac{D_1d_1^5}{rH}$$

Whence $d_1 = dr^{\frac{2}{5}}$, hence the following rule to determine stack dimensions for any altitude: Divide the barometric pressure at sea level (=30″) by the barometric pressure at the given altitude, and call the quotient r. Determine the stack height and diameter required at sea level, then multiply the height so determined by r, and the diameter by $r^{\frac{2}{5}}$, and the resulting dimensions apply at the given altitude. The flue area can be determined in the same way.

Table 52 gives values of r and $r^{\frac{2}{5}}$ computed from data in Table 12.* These show that altitude affects the height more than the area, and that practically no increase of area is needed for altitudes up to 3,000 feet. The grate areas should be increased in the same proportion as the stack areas

TABLE 52

Altitude in Feet above Sea Level.	Atmospheric Pressure. Lbs. per Sq.In.	r	$r^{\frac{2}{5}}$
15,221	8.19	1.80	1.27
14,075	8.56	1.72	1.24
12,934	8.94	1.65	1.22
11,799	9.33	1.58	1.20
10,127	9.95	1.48	1.17
9,031	10.38	1.42	1.15
7,932	10.82	1.36	1.13
6,843	11.28	1.30	1.11
5,764	11.76	1.25	1.09
5,225	12.01	1.22	1.08
4,169	12.51	1.18	1.07
3,115	13.03	1.13	1.05
2,063	13.57	1.08	1.03
1,539	13.84	1.06	1.02

Kent's Table—Mr. William Kent has prepared a table giving the size of boiler chimneys that has met with much approval.

TABLE 53

DIMENSIONS OF CHIMNEYS BY KENT'S FORMULA

Diameter in Inches.	Area in Square Feet.	HEIGHT OF CHIMNEY IN FEET.																									Equivalent Square Chimney. Side of Square. Inches.	Diameter in Inches.
		80	85	90	95	100	105	110	115	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200		
		COMMERCIAL HORSE POWER.																										
30	4.91	107	110																								27	30
33	5.94	133	137																								30	33
36	7.07	163	168	173																							32	36
39	8.30	196	202	208	214																						35	39
42	9.62	231	238	245	251	258	265	271																			38	42
48	12.57	311	320	330	338	348	356	365	373	381	389	396															43	48
54	15.90		415	427	438	449	461	472	482	493	503	513	523	532	542												48	54
60	19.64			536	551	565	579	593	606	619	632	644	657	669	680	692	704	715									54	60
66	23.76				678	694	711	728	744	760	776	791	806	821	835	849	864	877	891	904	918						59	66
72	28.27					835	856	876	896	915	934	952	970	988	1006	1022	1040	1056	1072	1089	1105	1120	1136	1151			64	72
78	33.18						1014	1038	1062	1084	1107	1129	1150	1171	1192	1212	1232	1252	1272	1291	1310	1328	1346	1364	1382	1400	70	78
84	38.48							1214	1241	1268	1294	1320	1345	1370	1394	1418	1441	1464	1487	1509	1531	1553	1574	1595	1616	1637	75	84
90	44.18								1435	1466	1496	1526	1555	1584	1612	1639	1666	1693	1719	1745	1771	1796	1820	1845	1869	1893	80	90
96	50.27								1643	1678	1713	1747	1780	1813	1845	1876	1907	1938	1968	1998	2027	2056	2084	2112	2140	2167	86	96
102	56.75									1905	1944	1983	2021	2058	2094	2130	2165	2200	2234	2268	2301	2333	2366	2397	2429	2460	91	102
108	63.62										2190	2234	2276	2318	2359	2399	2439	2478	2516	2554	2592	2628	2664	2700	2736	2771	96	108
114	70.88											2499	2547	2594	2640	2685	2729	2773	2816	2858	2900	2941	2982	3022	3061	3100	101	114
120	78.54												2833	2885	2936	2986	3036	3084	3132	3179	3226	3271	3316	3361	3405	3448	107	120
132	95.03												3450	3514	3576	3637	3697	3756	3815	3872	3929	3984	4039	4093	4147	4200	117	132
144	113.10													4205	4279	4352	4424	4495	4565	4632	4701	4768	4834	4898	4963	5026	128	144

*See page 58.

For ordinary rates of combustion of bituminous coals it is reliable, provided no unusual conditions are encountered. For ready reference in cases where approximate dimensions of a chimney based on boiler horse-power are required Mr. Kent's table is extremely convenient. In this the figures correspond to a coal consumption of 5 lbs. of coal per H. P. per hour. Mr. Kent says: "This liberal allowance is made to cover the contingencies of poor coal being used, and of the boilers being driven beyond their rated capacity. In large plants with economical boilers and engines, good fuel and other favorable conditions, which will reduce the maximum rate of coal consumption at any one time to less than 5 lbs. per H. P. per hour, the figures in the table may be multiplied by the ratio of 5 to the maximum expected coal consumption per H. P. per hour. Thus, with conditions which made the maximum coal consumption only 2.5 lbs. per hour, the chimney 300 ft. high $\times$ 12 ft. diameter should be sufficient for $6155 \times 2 = 12{,}310$ horse power.

STACKS FOR BOILERS USING OIL FUEL

The requirements for stacks attached to boilers burning oil are entirely different from those required when burning coal. The loss of draft caused by a bed of coals is eliminated, the volume of flue-gas will be less than for an equal weight of coal if the air supply is properly adjusted, and the action of the burner is in a measure equivalent to a forced draft. Experimental data such as are available for coal have not been gathered for oil, hence no such elaborate methods of determining proportions of stacks for oil as have been worked out for coal, are at present available, but the method to be given has been found entirely satisfactory for a large number of cases, and may be used without hesitation.

A stack 75 to 80 feet high above a boiler damper plate furnishes ample draft to burn oil, if there are no long flues or turns in the breeching, hence the only other requirement is to determine the diameter. Owing to the smaller volume of gas formed as compared with coal, and the forced draft action of the burner, it has been found by experience that when oil is burned, a stack having 60 per cent. of the cross section required by the same boiler if bituminous coal were used, will be amply large to enable the boiler to be fired at 50 per cent. above rating with oil. Example: required the dimensions of a stack for 500 H. P. of boilers burning oil. $500 \times .60 = 300$. Kent's Table may be used with facility in this case, hence referring to this it will be found that a stack 48 inches in diameter and 80 feet high will develop 311 H. P. with bituminous coal, hence a 48-inch stack will meet the requirements of this case. Many plants are operating successfully with stack areas equal to only 50 per cent. of the coal area, but this allowance is too scant to provide properly for overloads.

Correction for Flues—It has already been shown that a flue 100 feet long loses 0.1 inch of draft, and that a right-angled turn loses .05 inch. But Table No. 51 shows that for a stack temperature of 450° and external air temperature of 80° the draft in a 100-foot stack will be 0.549 inches, hence the draft due to one foot of height will be practically 0.0055 inches, consequently for each elbow in the breeching an addition of 10 feet to the height of the stack will be needed, and for a length of flue 100 feet long, an addition of $0.1 \div 0.0055 = 18.1$ feet, will be sufficient.

Where local conditions, such as buildings, etc., necessitate use of stacks exceeding 80 feet in height the corresponding diameter may be found in the same way. Example: if for 1000 H. P. a stack 140 feet high were assumed, and the breeching were 50 feet long and contained two right-angled turns, the part of the height required to give the additional draft for flue and turns would be $2 \times 10 + 9 = 29$ feet. $140 - 29 = 111$ feet. $1000 \times .60 = 600$ H. P. From Table 53 a stack 60 inches diameter, and 110 feet high, the nearest tabular value to 111, is equal to 593 H. P., hence a 60-inch stack would be suitable for the given conditions.

DRAFT GAUGES

The ordinary form of draft gauge consisting of the U-tube, Fig. 39, containing water, lacks sensitiveness when used for measuring

such slight pressure differences as exist in a chimney, hence gauges which multiply the draft indications are more convenient, and are much used.

Barrus' Gauge—Mr. G. H. Barrus for a number of years has used with excellent results an instrument which multiplies the ordinary indications as many times as is de. sired. It is illustrated in Fig. 40, and consists of a U-tube made of ½-inch glass, surmounted by two larger tubes, or chambers, each having a diameter of 2½-inch. Two different liquids which will not mix, and which are of different color, are used. The movement of the line of demarcation is proportional to the difference in the areas of the chambers and of the U-tube connecting them below. The liquids generally employed are alcohol colored

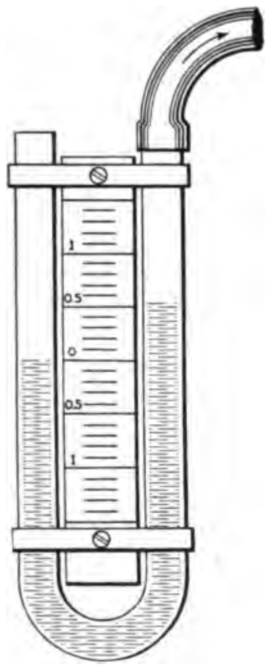


FIG. 39

U-TUBE DRAFT GAUGE

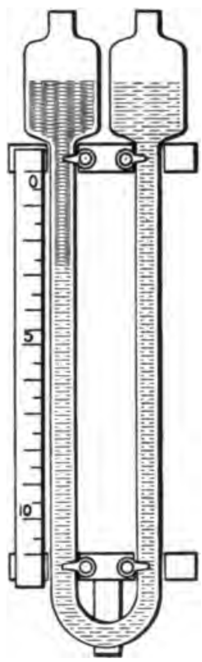


FIG. 40

BARRUS' DRAFT GAUGE

red and a certain grade of lubricating oil. A multiplication varying from eight to ten times is obtained under these circumstances; in other words, with ¼-inch draft the movement of the line of demarcation is some 2 inches. The instrument is calibrated by referring it to the ordinary U-tube gauge.

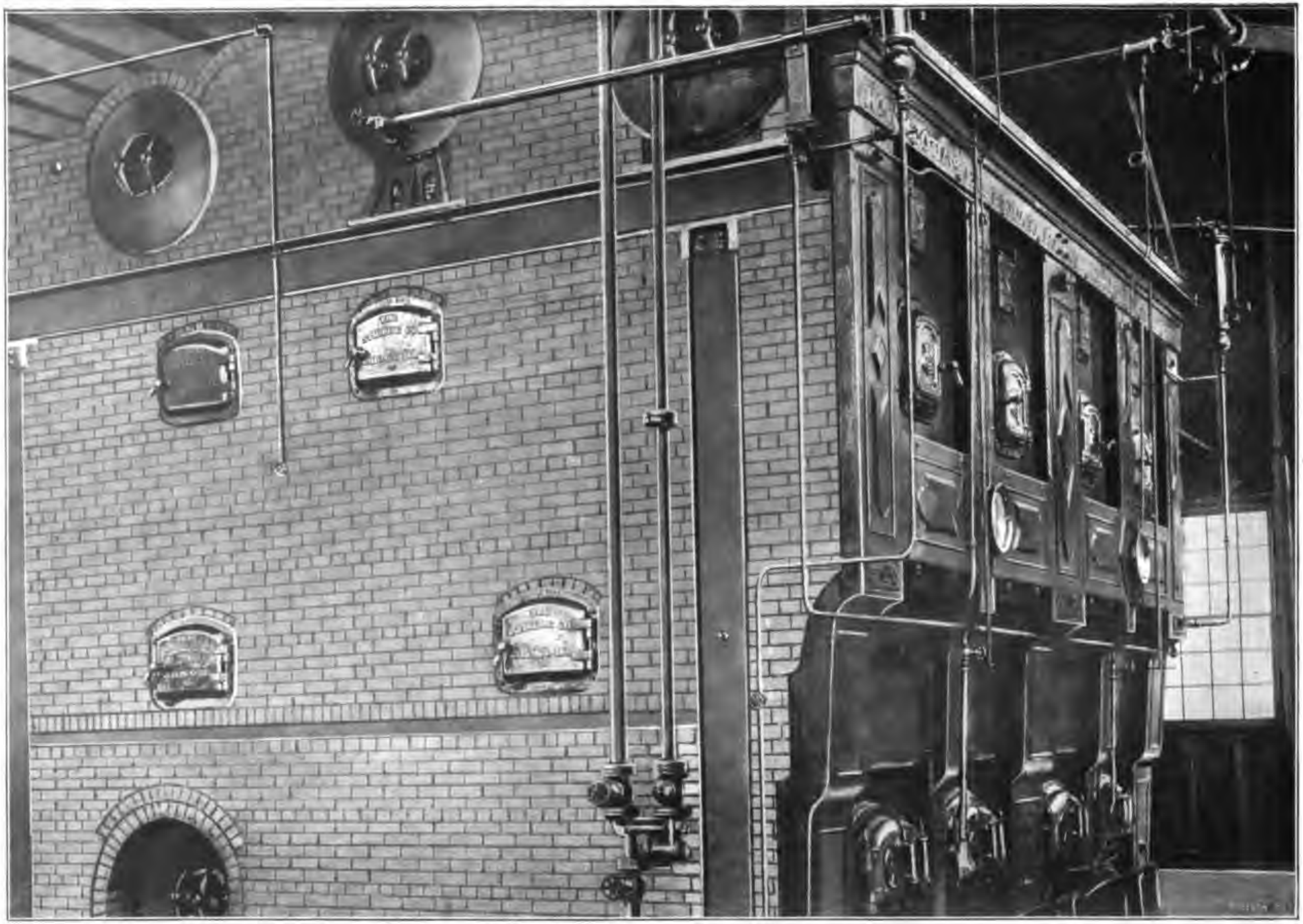
Ellison's Gauge—In this form of gauge the lower portion of the ordinary U-tube has been replaced by a tube slightly inclined to the horizontal, as shown in Fig 41. By this arrangement any vertical motion in the left hand upright tube causes a very much greater travel of the liquid in the inclined tube, thus permitting extremely small variation in the draft pressure to be read with facility.

FIG. 41. ELLISON'S DRAFT GAUGE

The gauge is first leveled by means of the small level attached to it, both legs being open to the atmosphere. The liquid is then adjusted (by adding to or taking from it) until its meniscus rests at the zero point on the right. The left hand leg is then connected to the source of draft by means of a piece of rubber tubing. Under these circumstances, a rise of level of one inch in the left hand vertical tube causes the meniscus in the inclined tube to pass from the point 0 to 1.0. The scale is divided into tenths of an inch, and the subdivisions are hundredths of an inch.

The right hand leg of the instrument bears two marks. By filling the tube to the lower of these the range of the instrument is increased one-half inch, *i. e.* it will record draft pressures from 0 to 1½ inches. Similarly, by filling to the upper mark, the range is increased to 2 inches. When so used the observed readings in the scale are to be increased by one-half or one-inch, as the case may be.

The makers recommend the use of a non-drying oil for the liquid, usually a 300° test refined petroleum, but water suffices for all practical purposes.



PART OF 2.700 H. P. OF STIRLING BOILERS, NOVA SCOTIA STEEL & COAL CO., NORTH SIDNEY, N. S.

Analysis of Flue-Gases

In the chapter on Combustion* the effect of excess air in cooling the fire is set forth. This excess air would not reduce the boiler efficiency if the gases of combustion, when sweeping over the heating surface, were cooled to the initial temperature of the air supply, since before their exit they would give up the heat they absorbed after entering the furnace. Such abstraction of the heat is not possible in a boiler because the flue-gases cannot be reduced to a temperature below 50° to 100° above the temperature of the steam. With a fixed temperature of discharge the loss in the waste gases is proportional to the weight of the gases, hence excess air not only reduces the temperature of the furnace, which causes a decrease in the boiler capacity and efficiency but it causes still further loss by serving as a vehicle to convey heat to the stack.

An insufficient air supply causes formation of carbon monoxide (CO) instead of carbon dioxide (CO_2), and if this passes away unburned the heat derived from a pound of carbon will be only 4450 B. T. U. instead of the 14600 B. T. U. obtainable when carbon dioxide is formed.

If the combustion were perfect and no excess air were admitted, the resulting gases would be carbon dioxide, and steam, together with the nitrogen from the air. The amount of carbon dioxide and nitrogen would bear a fixed ratio to the carbon burned. Consequently, since some excess air is unavoidable, the nitrogen in the flue-gases furnishes an index to the amount of that excess, and the presence of carbon dioxide indicates incomplete combustion of carbon.

Object of the Analysis—The object of the flue-gas analysis is to determine from a sample of the gas the amount of excess air admitted, the degree of completeness of the combustion of the carbon, and the amount and distribution of the heat losses due to the excess air and incomplete combustion. The quantities actually determined by the analysis are the relative proportions of carbon dioxide (CO_2), carbon monoxide (CO), and oxygen (O) in the gases. Although the analysis does not directly determine the amount of nitrogen present in the flue-gases, yet its actual amount, as well as that of the air supply, may readily be ascertained by calculation. When air is drawn through an opening, like an ash-pit door, sometimes an anemometer can be used for ascertaining the velocity through the area, and the air supply be determined by these means.

Before describing in detail the apparatus and methods used for analyzing flue-gases, the application of the results obtainable from the analysis will be illustrated.

A pound of carbon requires for complete combustion, 2.67 pounds of oxygen, or a volume of 32 cubic feet at 60° F., and the gaseous product, carbon dioxide (CO_2), when cooled, occupies precisely the same volume as the oxygen, viz., 32 cubic feet. If the oxygen is mixed with nitrogen in the same proportion as it is found in air (20.91 O and 79.09 N), the volume of the carbon dioxide (CO_2) after combustion, and also its proportion to nitrogen, is the same as that of the oxygen; hence, for complete combustion of carbon, with no excess of air, the volumetric analysis of the flue gases is,

Carbon dioxide . .	CO_2=20.91%
Carbon monoxide .	CO =None
Oxygen	O =None
Nitrogen. . . .	N =79.09%

If the supply of air is in excess of that required to supply the oxygen needed, the combined volumes of the carbon dioxide and oxygen are still the same as that of the oxygen before combustion; consequently, *for the complete combustion of pure carbon, the sum of the percentages by volume of the carbon dioxide and oxygen in the flue gases must always be 20.91, no matter what the supply of air may be.*

Carbon monoxide (CO) produced by imperfect combustion of carbon, occupies *twice* the volume of the oxygen entering into its composition, and renders the volume of the flue gases greater than that of the air supply in the proportion of

$$\frac{100}{100-\frac{1}{2}\text{ the \% of CO}}, \text{ hence}$$

*Article "Temperature of the Fire," page 107.

when pure carbon is the fuel, the sum of the percentages of carbon dioxide, oxygen, and one-half the carbon monoxide, must be in the same ratio to the nitrogen as is oxygen in air, i.e., 20.91 to 79.09

The action of hydrogen in coal is to increase the apparent percentage of nitrogen in the flue-gases, because the water vapor condenses at the temperature at which the analysis is made, and account of it is lost, but the nitrogen that accompanied the oxygen with which the hydrogen combined, maintains its gaseous form and passes into the analyzing apparatus with the other gases.

Example—Suppose an analysis of flue-gases shows 12.5% of carbon dioxide, 0.6% of carbon monoxide and 6.5% of oxygen, all by volume. Then nitrogen, which is the only other constituent in the flue-gases worthy of consideration, will represent a percentage of the total volume,

$$100-(12.5+0.6+6.5)=80.4\%$$

Assume the unit of volume here designated as 100% to represent 100 cubic feet. From Table 54, the weights of the various gases per cubic foot are as follows:

Carbon dioxide	(CO_2)	=0.122681
Carbon monoxide	(CO)	=0.078071
Oxygen	(O)	=0.088843
Nitrogen	(N)	=0.078314

The weight of the flue-gas per unit volume of 100 cubic feet will therefore be

Carbon dioxide	(CO_2)	$=.12268\times 12.5=1.5534$ lbs.
Carbon monoxide	(CO)	$=.07807\times 0.6=.0468$
Oxygen . .	(O)	$=.08884\times 6.5=.5775$
Nitrogen . .	(N)	$=.07831\times 80.4=6.2961$
		100.0 8.4738 lbs.

From the atomic or combining weights of the elements, it is known that in a unit of carbon dioxide, oxygen constitutes eight-elevenths of the weight, the remaining three-elevenths being carbon; and in carbon monoxide four-sevenths is oxygen by weight, and three-sevenths carbon. Therefore, the weight of oxygen in 100 cubic feet of flue-gas in question is.

Oxygen in CO_2 . .	$=1.5534\times 8/11=1.1297$ lbs.
Oxygen in CO . .	$=0.0468\times 4/7=0.0267$
Free Oxygen . . . = ...	$=0.5775$
Total weight of Oxygen . . .	$=1.7339$ lbs.

The weight of carbon as determined from the same gas analysis is,

Carbon in CO_2 . .	$=1.5534\times 3/11=0.4236$ lbs.
Carbon in CO . .	$=.0468\times 3/7=0.0201$
Total weight of Carbon . . .	$=0.4437$ lbs.

As atmospheric air supplied to the fire contains 23.15% of oxygen by weight, then the weight of air which contained 1.7339 lbs. of oxygen is $\frac{1.7339\times 100}{23.15}=7.49$ lbs; and, as this amount of air was required for the combustion of 0.4437 lbs. of carbon, the weight of air per pound of carbon is

$$\frac{7.49}{0.4437}=16.88 \text{ lbs.}$$

TABLE 54

DENSITY OF GASES AT ATMOSPHERIC PRESSURE

*(Adapted from Kent.**)*

GAS.	SYMBOL.	Specific Gravity Air = 1.	Weight of One Cubic Foot at 32° F. Pounds.	Cubic Feet per Pound at 32° F.	Relative Density, Hydrogen = 1	
					Exact Relative Densities	Approximate Whole Numbers.
Oxygen	O	1.10521	0.088843	11.257	15.96	16
Nitrogen . . .	N	0.9701	0.078314	12.764	14.01	14
Hydrogen . . .	H	0.069234	0.005589	178.930	1.00	1
Carbon dioxide .	CO_2	1.51968	0.122681	8.158	21.95	22
Carbon monoxide	CO	0.96709	0.078071	12.818	13.97	14
Methane . . .	CH_4	0.55297	0.044640	22.412	7.99	8
Ethylene . . .	C_2H_4	0.96744	0.078100	12.804	13.97	14
Acetylene . . .	C_2H_2	0.89820	0.073010	13.697	12.97	13
Sulphur dioxide .	SO_2	2.21295	0.178646	5.598	31.96	32
Air	....	1.0000	0.080728	12.383	.....	..

**Steam Boiler Economy*, p. 20.

If the coal in the example considered contained 86% of carbon, 4% of hydrogen, and 2.5% of oxygen, then the air per pound of coal=16.88×.86=14.52 lbs., disregarding the hydrogen and oxygen. But the oxygen in fuel renders inert $\frac{1}{8}$ of its weight of hydrogen, and only the remainder of the hydrogen is available for combustion; therefore the air required to burn the hydrogen is

$$\left\{ .04 - \frac{.025}{8} \right\} 34.56 = .0369 \times 34.56^{*} = 1.275 \text{ lbs.}$$

Thus the total air supply per pound of fuel becomes 14.52+1.28=15.80 lbs.

Air Required and Supplied—When the ultimate analysis of a fuel is known the air required for complete combustion, with no excess, can be found as shown in chapter on Combustion† or from the following approximate formula:

Pounds of air required per pound of fuel=

$$34.56 \left\{ \frac{C}{3} + H - \frac{O}{8} \right\} \qquad [43]$$

where C, H and O=per cent. by weight of carbon, hydrogen and oxygen in the fuel, divided by 100.

When the flue-gas analysis is known, the total amount of air supplied is,‡

Pounds of air supplied per pound of fuel=

$$3.032 \left\{ \frac{N}{CO_2+CO} \right\} \times C \qquad [44]$$

where N, CO and CO_2=percentage by volume of nitrogen, carbon monoxide and carbon dioxide in the flue-gases, and C the proportionate part, by weight, of carbon in the fuel.

The weight of the flue-gases will be one minus the per cent. of ash (expressed in hundredths) more than this, *i. e.*, it will be the sum of the weights of the air, and the combustible and moisture in the fuel, hence Weight of flue-gases per lb. of fuel=

$$3.032 \left\{ \frac{N}{CO_2 + CO} \right\} \times C + (1 - A) \qquad [46]$$

where A=proportionate part, by weight, of ash in the fuel.

The ratio of the air actually supplied per pound of *carbon* to that theoretically required to burn it is

$$3.032 \frac{N}{CO_2 + CO} \div 11.52$$

$$= 0.2632 \frac{N}{CO_2 + CO} \qquad [47]$$

in which N, CO_2 and CO are the percentages by volume in the flue-gas.

The ratio of the air supplied per pound of *fuel* to the amount theoretically required is

$$\frac{N}{N - 3.782\, O} \qquad [48]$$

which is derived as follows: The N in the flue-gas is the content of nitrogen in the whole amount of air furnished. The oxygen in the flue-gas is due to the air supplied and not used. This oxygen was accompanied by 3.782 times its volume of nitrogen. $(N-3.782\, O)$ represents the nitrogen content in the air actually required for combustion. Hence $N \div (N-3.782\, O)$ is the ratio of the air supplied to that required. The per cent. of excess air is this ratio minus one. Table 55 gives the values of this ratio corresponding to various percentages of CO_2+CO and CO_2+CO+O

The heat lost in the flue-gases is

$$L = 0.24\, W(T - t) \qquad [49]$$

where

L = B. T. U. lost per pound of fuel.

W = Weight of flue-gases in pounds per pound of fuel.

T = Temperature of flue-gases.

t = Temperature of atmosphere.

0.24 is the specific heat of flue-gas.

The heat lost in the carbon monoxide in B. T. U. per pound of fuel is

$$L' = 10150 \times \left(\frac{CO}{CO + CO_2}\right) \times C \qquad [50]$$

where, as before, CO and CO_2 are the per cent. by volume in the flue-gas, and C the proportion (by weight) of carbon in the fuel.

---

*Weight of air required for the combustion of one pound of hydrogen. †Article, "Air Required," p. 106. ‡The derivation of this formula may be found in Kent's *Steam Boiler Economy*, First Edition, page 32. As a check the following formula may be used:

$$\text{Pounds of air supplied per pound of fuel} = 11.52 \times \frac{CO_2 + \frac{1}{2}CO + O}{CO_2 + CO} \times C + 34.56 H^1 \qquad [45]$$

Where H^1 is the available hydrogen $(H - \frac{1}{8}O)$ in the fuel. This formula and that above given will not produce the same result unless the flue-gas and coal analyses are accurate, and the sample of the gas is a true one. The more accurate the work the more nearly the formulas will agree.

Orsat Apparatus—The analysis of the flue-gases is best made for practical purposes by means of the Orsat apparatus, shown in Fig. 42. The operation is as follows: Exactly 100 cc of the gas sample are drawn into the graduated measuring burette, *A*, and then passed in succession into the U-form absorbing vessels, *D*, *E*, *F*, each time being returned to and measured in *A*. In passing into the U-shaped vessels, the gas displaces the liquid contained therein, driving it up into the other legs. A portion of the fluids, however, adheres to the glass tubes placed in the vessels for that purpose, and comes in intimate contact with the gas. Each vessel absorbs a different constituent. *D* is filled with a solution of potassium hydroxide and takes up the carbon dioxide; *E* contains pyrogallic acid,* which removes the oxygen; and *F* absorbs the carbon monoxide in a solution of cuprous chloride. The reduction in volume measured in *A* gives the percentage of each constituent gas.

The connections to *A* are made through the glass stop cocks *M* and the capillary

TABLE 55

RATIO OF TOTAL AIR SUPPLIED TO THAT THEORETICALLY REQUIRED FOR VARIOUS ANALYSES OF FLUE-GASES

$$\text{RATIO} = \frac{N}{N - 3.782\,O}$$

CO_2+CO	N = 79 CO_2+CO+O = 21	N = 79.5 CO_2+CO+O = 20.5	N = 80 CO_2+CO+O = 20	N = 80.5 CO_2+CO+O = 19.5	N = 81 CO_2+CO+O = 19	N = 81.5 CO_2+CO+O = 18.5	N = 82 CO_2+CO+O = 18
21	1.00	....	....	....	....	....	....
20	1.05	1.02	1.00	....	....	....	....
19	1.11	1.08	1.05	1.02	1.00	....	....
18	1.17	1.14	1.10	1.08	1.05	1.02	1.00
17	1.24	1.20	1.17	1.13	1.10	1.07	1.05
16	1.32	1.27	1.23	1.20	1.16	1.13	1.10
15	1.40	1.35	1.31	1.27	1.23	1.19	1.16
14	1.51	1.45	1.39	1.35	1.30	1.26	1.23
13	1.62	1.55	1.50	1.44	1.39	1.34	1.30
12	1.76	1.68	1.61	1.54	1.49	1.43	1.38
11	1.92	1.82	1.74	1.66	1.60	1.53	1.48
10	2.11	2.00	1.90	1.81	1.72	1.65	1.59
9	2.35	2.21	2.08	1.97	1.88	1.79	1.71
8	2.65	2.47	2.31	2.18	2.06	1.95	1.86
7	3.03	2.80	2.59	2.44	2.27	2.14	2.03
6	3.55	3.22	2.96	2.74	2.54	2.38	2.24
5	4.27	3.81	3.44	3.14	2.89	2.68	2.50
4	5.37	4.65	4.11	3.68	3.34	3.05	2.83
3	7.23	5.97	5.10	4.45	3.96	3.56	3.25
2	11.06	8.34	6.71	5.63	4.85	4.27	3.82
1	23.51	13.83	9.83	7.64	6.27	6.12	4.64

*SOLUTIONS FOR ORSAT APPARATUS

For absorbing CO_2—*Caustic Potash.* Dissolve one part by weight of caustic potash in two and one-half parts of water.

For absorbing *O*—*Pyrogallol.* Dissolve one part by weight of pyrogallic acid in two parts of hot water, and three parts of caustic potash solution, made as above directed.

For absorbing CO—*Cuprous Chloride.* Dissolve one part by weight of cuprous chloride in seven parts of hydrochloric acid, then add two parts of copper clippings and let stand for twenty-four hours, afterwards adding three parts of water before use.

tube C. The movement of the gases is produced by lowering or raising the bottle L, which is connected to the lower part of A by the rubber tube S, and is partially filled with water. When a measurement is taken, the level of the water in A and L must be the same, so that all measurements are taken at atmospheric pressure. A constant temperature of the gas in A is maintained by the water in the surrounding cylinder shown.

The sample is drawn into the apparatus through the cock B, which also serves to connect the capillary tube to the atmosphere, the latter connection being through the spindle of the cock; this permits the removal of any excess of gas above 100 cc that may have been drawn into A. Before the sample is drawn, the vessels D, E and F should have their respective liquids raised to the cocks M (which can then be closed, and the atmospheric pressure acting through the other leg, which is open, will keep them filled); the burette A and the capillary tubes should be filled with water up to the cock B. All this can easily and quickly be done by raising and lowering L, and opening and closing cocks M and B. The absorption of oxygen and carbon monoxide is very slow, and the gas should be passed back and forth a number of times until a reduction of volume is no longer indicated.

As the pressure of the gases in a flue is less than the atmospheric pressure, they will not, of themselves, flow through the rubber or metal tubing connecting to the analyzing apparatus; but by filling the instrument two or three times and discharging it into the atmosphere through cock B, the air can be removed from the connecting tubing and a sample of the gas be obtained. For rapid work, an aspirator can be used for drawing the gas from the tube in a constant stream. If this is used there is less danger of an admixture of air. It is sometimes desirable to take a sample that represents an average during half an hour or an hour, and in this case a metal or glass vessel with a stop-cock at both top and bottom, and filled with water, can be connected through the upper stop-cock to the flue, and the bottom cock then be opened. The water will gradually drip out, drawing the gas into the vessel. The time taken to fill it can be regulated by the lower cock.

The result of a flue-gas analysis depends both on the manner and time of taking the sample, and to get at the average composition of the gas, a number of determinations should be made on samples from different parts of the flue.

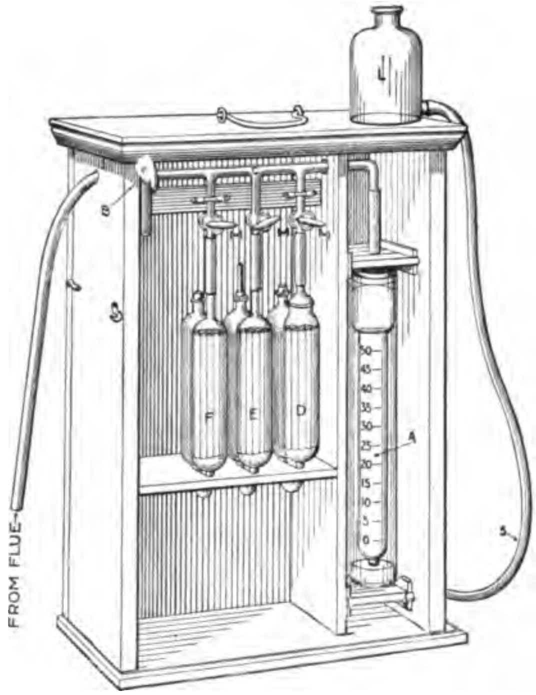


FIG. 42. ORSAT APPARATUS FOR FLUE-GAS ANALYSIS

The analysis made by the Orsat apparatus is volumetric; if the analysis by weight is required it can be found from the volumetric analysis as follows:

Multiply the percentages by volume by the molecular weight of the gas, and divide by the sum of all the products; the quotient will be the percentage by weight.

The molecular weights are as follows:

Carbon dioxide	44
Carbon monoxide	28
Oxygen	32
Nitrogen	28

Application of Formulas and Rules—Pocahontas coal used in a furnace was composed of 82.1 % Carbon
4.25% Hydrogen
2.6 % Oxygen
6. % Ash

The flue-gas analysis gave

Carbon dioxide	CO_2	$=10.6\%$
Oxygen	O	$=10.$
Carbon monoxide . . .	CO	$=0.$
Nitrogen (by difference) .	N	$=79.4$

Determine:—The flue-gas analysis by weight, the amount of air required for perfect combustion, the actual weight of air per pound of coal, the weight of flue-gas per pound of coal, the heat loss in chimney if gases are discharged at 500° F., and the ratio of the air supplied per pound of coal to that theoretically required. Solution:

Weight of air for perfect combustion=

$$34.56\times\left\{\frac{.821}{3}+.0425-\frac{.026}{8}\right\}=10.81 \text{ lbs.}$$

(Formula 43)

Actual weight of air per pound of coal=

$$3.032\times\frac{.794}{.106-.00}\times .821=18.64 \text{ lbs.}$$

(Formula 44)

Weight of flue-gases per pound of coal= $18.64+1-.06=19.58$ lbs. (Formula 46)

Heat lost in flue-gases per pound of coal= $.24\times 19.58\times(500-60)=2063.25$ B. T. U.'s. (Formula 49)

The coal contains about 14,500 B. T. U.'s, so there is $\frac{2,063.25}{14,500}=14.2\%$ of the heat lost in the flue-gases.

Ratio of air supplied to that required=

$$\frac{.794}{.794-.10\times 3.782}=1.90 \text{ (Formula 48.)}$$

This may also be calculated from the first two results above, $\frac{18.64}{10.81}=1.72$, the difference between this ratio and that obtained from Formula [48] being due to inaccuracies in either the flue-gas or coal analysis, or in both.

Table 56 shows the method of converting a flue-gas analysis by volume into an analysis by weight.

TABLE 56

ANALYSIS OF FLUE-GASES

GAS.	Analysis by Volume.	Molecular Weight.	Volume × Molecular Weight.	Analysis by Weight.
Carbon dioxide . . (CO_2)	10.6	$12+2\times16$	466.4	$\frac{466.4}{3.009.6}=15.5\%$
Carbon monoxide . . (CO)	0.0	$12+16$	0.0	$\frac{0.0}{3009.6}=0.0\%$
Oxygen. . . . . (O)	10.0	2×16	320.0	$\frac{320.0}{3009.6}=10.7\%$
Nitrogen . . . . (N)	79.4	2×14	2223.2	$\frac{2223.2}{3009.6}=73.8\%$
		Total . .	. 3009.6	

Steam Boiler Efficiency

The efficiency of a boiler is the ratio between the heat units utilized in production of steam, and the heat units contained in the fuel used. But whenever solid fuel such as coal is used, it is impossible to prevent a portion of it from falling through the grates, where it mixes with the ashes without burning, and generates no heat. The boiler itself cannot justly be charged with failure to absorb the heat value represented by the fuel wasted through the grates, but the boiler owner must pay for the fuel so wasted, and is justified in charging this waste to the combination of boiler and furnace. The heat supplied to the boiler is that due to the combustible actually burned, irrespective of how much may be dropped through the grates. In consequence *two* efficiencies may be determined, viz.:

(1) Efficiency of the boiler=

$$\frac{\text{Heat absorbed per lb. of combustible}}{\text{Heat value of one lb. of combustible*}} \qquad [51]$$

(2) Efficiency of boiler and grate=

$$\frac{\text{Heat absorbed per pound of fuel}}{\text{Heat value of one pound of fuel,}} \qquad [52]$$

The first is of value in comparing relative performances of boilers apart from the particular kind of grate used under them; the second is of value in comparing performances of different kinds of fuels, grates, etc., under the same boiler. If the loss of fuel through the grates could be wholly obviated, then the two efficiencies would be identical, as in case of a boiler fired with oil. Thus, if a coal contained 90% combustible, efficiency (1) would be

$$\frac{\text{Heat absorbed per lb. of fuel} \times .90}{\text{Heat value of one lb. of fuel} \times .90}$$

which reduces to efficiency (2). Similarly, efficiency (2) will in *any particular case* figure out the same whether the fuel be taken as *dry coal*, or coal as fired with its content of moisture. Example:—If the coal contained 3% of moisture, efficiency (2) would be

$$\frac{\text{Heat absorbed per pound of dry coal} \times 0.97}{\text{Heat value of one lb. of dry coal} \times 0.97}$$

Here the content of moisture cancels out, hence efficiency (2) may be based on either dry coal, or coal as actually fired.

Assume the following data:

Steam pressure by gauge . . .	149 lbs.
Temperature of feed water . . .	84° F.
Weight of coal as fired	7152 lbs.
Percentage of moisture in coal . .	.96
Total ash and refuse.	286 lbs.
Percentage of moisture in steam .	0.5
Total water evaporated	68664 lbs.

Analysis of the coal, by weight.

Moisture	0.96%
Ash	2.19
Carbon	87.76
Hydrogen	4.11
Oxygen, nitrogen and sulphur .	4.98
	100.00%

Heat value per pound of dry coal by calorimeter 15450 B. T. U.

The factor of evaporation for the conditions named is 1.182, hence the equivalent evaporation from and at 212° is 68664 × 1.182 = 81161 lbs., and, corrected for the moisture present, is .5% less, or 80755 lbs. of dry steam from and at 212°. This contains 80755 × 966 = 78,009,330 B. T. U.

The dry coal fired was

$$7152 - (.0096 \times 7152) = 7083 \text{ lbs.}$$

The ash-pit contained 286 lbs. of ash and refuse, whence the combustible, or coal dry and free from ash, was 7083−286=6797 lbs. The heat units in the steam are therefore:

Per pound of coal as fired,

$$\frac{78{,}009{,}330}{7152} = 10907 \text{ B. T. U.} \quad \text{(a)}$$

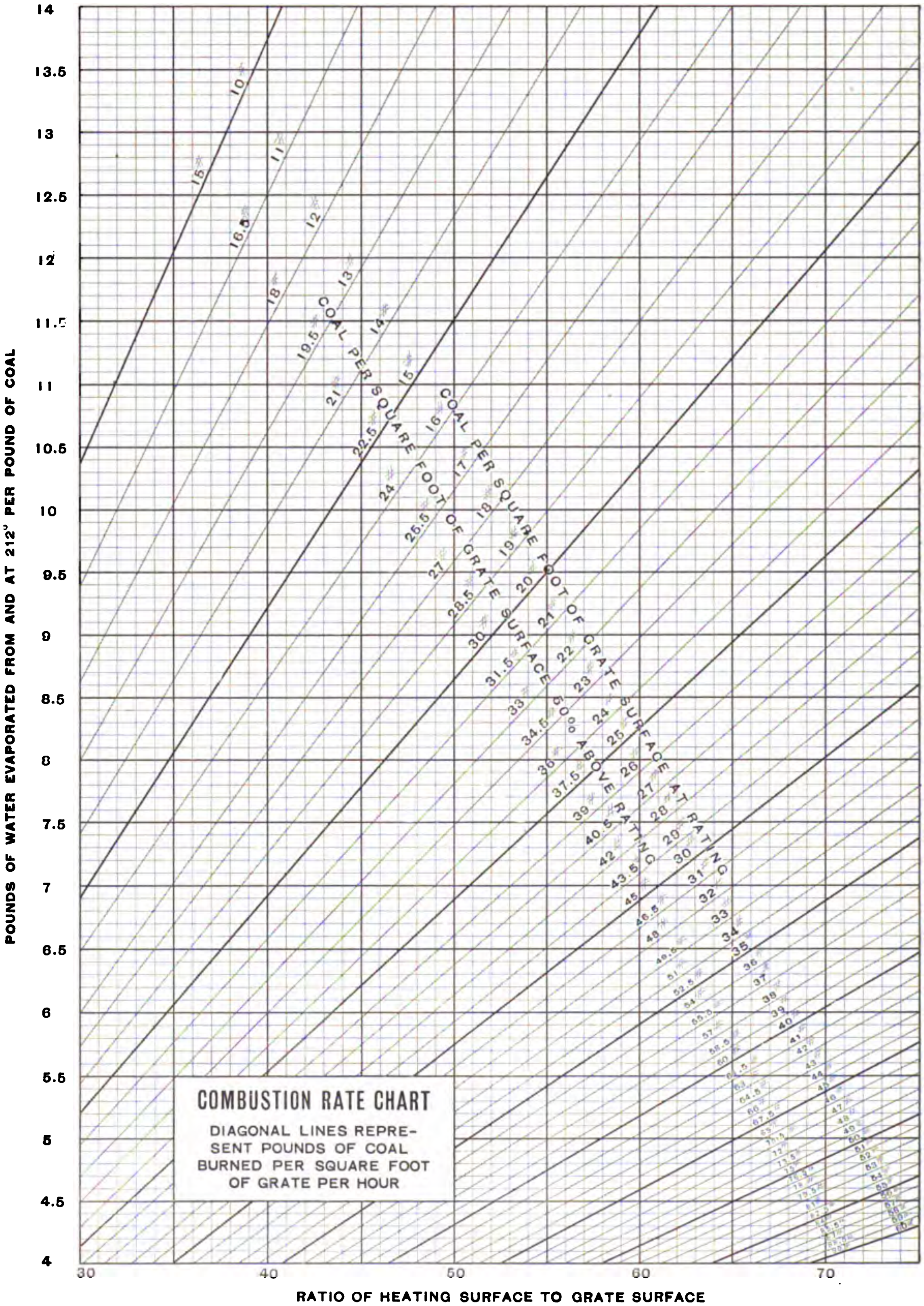
Per pound of dry coal,

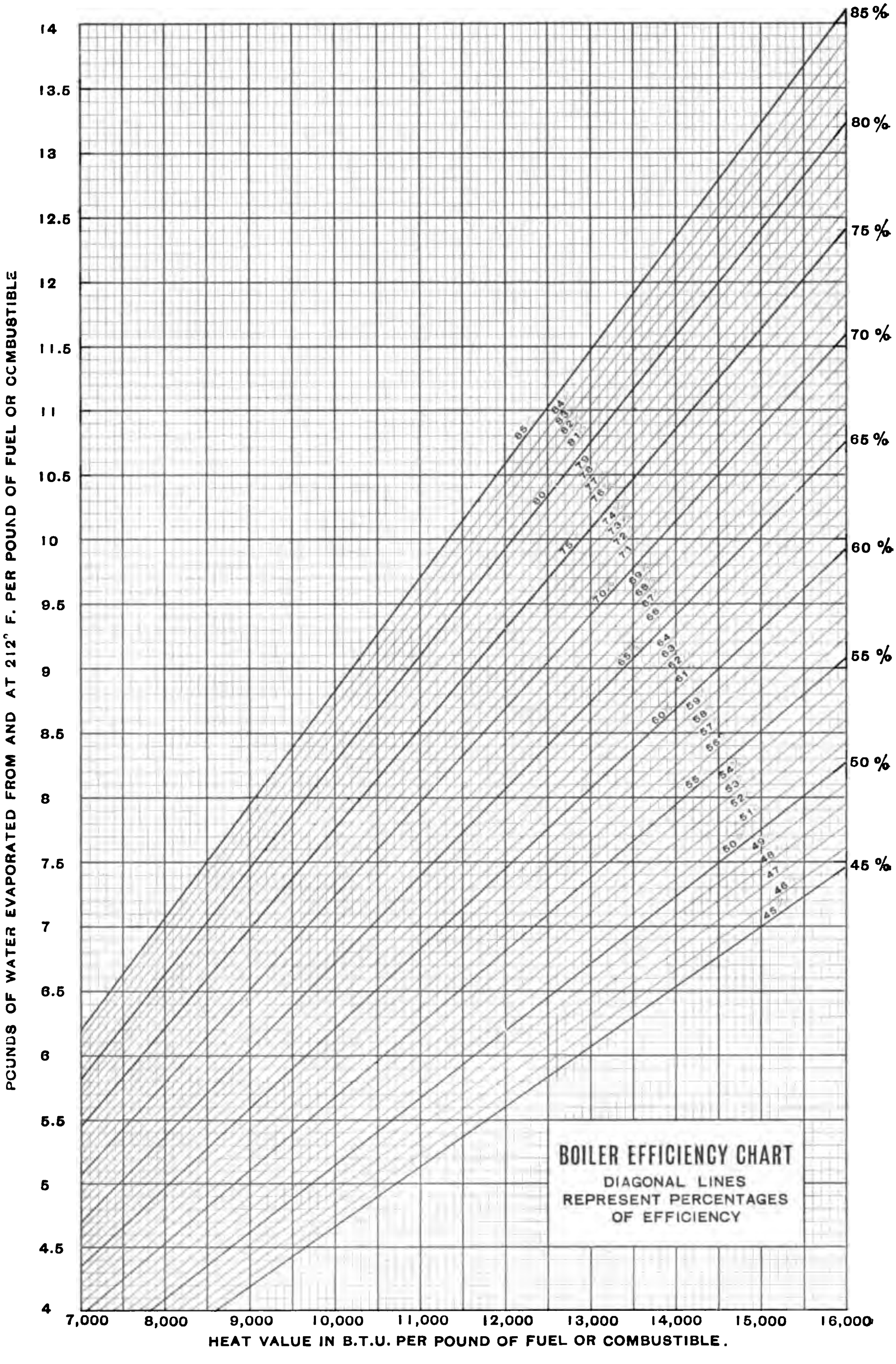
$$\frac{78{,}009{,}330}{7083} = 11013 \text{ B. T. U.} \quad \text{(b)}$$

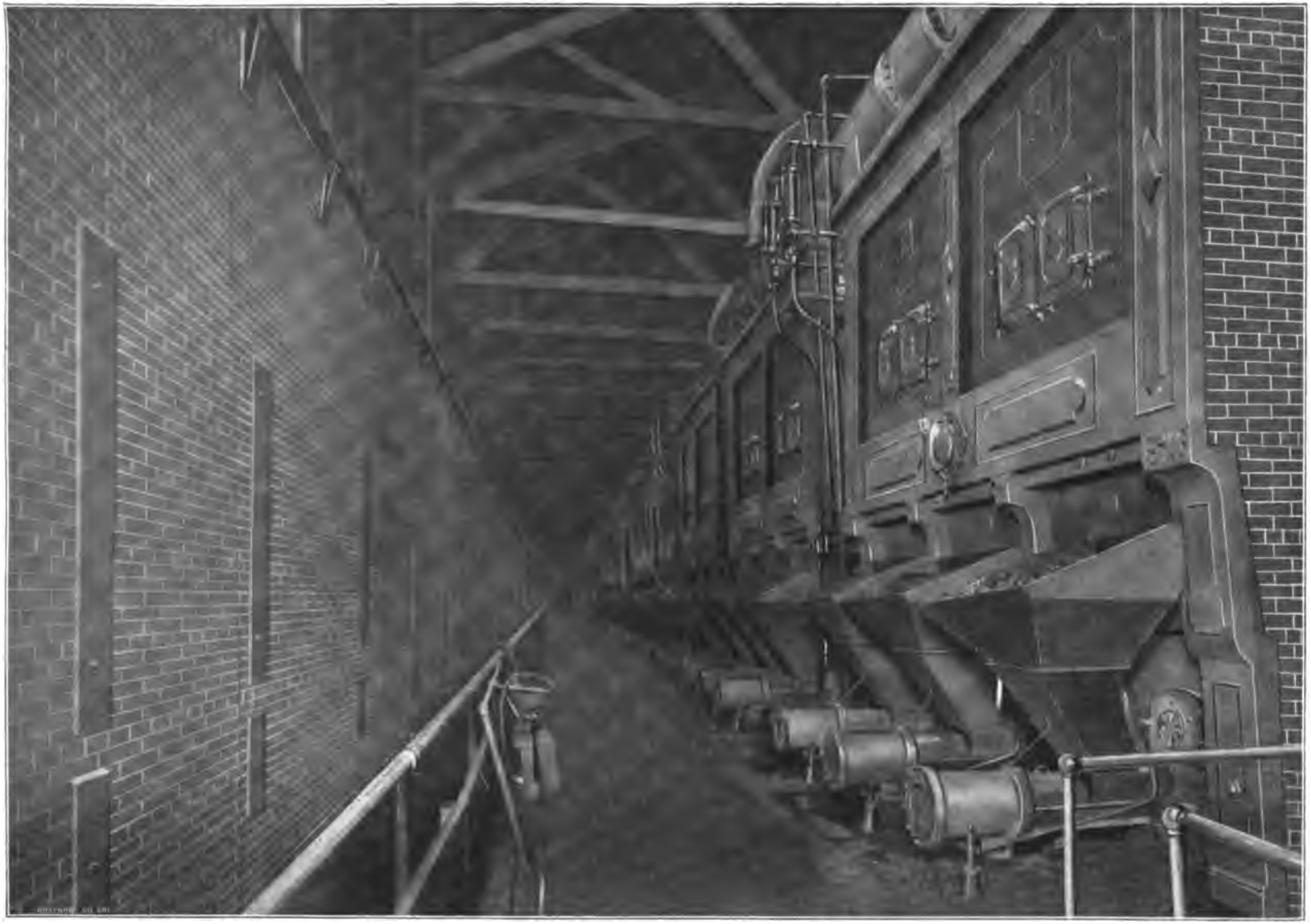
Per pound of combustible,

$$\frac{78{,}009{,}330}{6797} = 11477 \text{ B. T. U.} \quad \text{(c)}$$

*By combustible is here meant that part of the fuel *dry and free from ash*. Nitrogen and oxygen are thus included. Neither is combustible in strict accuracy, but custom has included them, as they form part of the volatile content of coal.


COMBUSTION RATE CHART
DIAGONAL LINES REPRESENT POUNDS OF COAL BURNED PER SQUARE FOOT OF GRATE PER HOUR
COAL PER SQUARE FOOT OF GRATE SURFACE
COAL PER SQUARE FOOT OF GRATE SURFACE AT RATING
COAL PER SQUARE FOOT OF GRATE SURFACE 50% ABOVE RATING
POUNDS OF WATER EVAPORATED FROM AND AT 212° PER POUND OF COAL
RATIO OF HEATING SURFACE TO GRATE SURFACE
14
13.5
13
12.5
12
11.5
11
10.5
10
9.5
9
8.5
8
7.5
7
6.5
6
5.5
5
4.5
4
30
40
50
60
70


BOILER EFFICIENCY CHART
DIAGONAL LINES
REPRESENT PERCENTAGES
OF EFFICIENCY
POUNDS OF WATER EVAPORATED FROM AND AT 212° F. PER POUND OF FUEL OR COMBUSTIBLE
HEAT VALUE IN B.T.U. PER POUND OF FUEL OR COMBUSTIBLE.
14
13.5
13
12.5
12
11.5
11
10.5
10
9.5
9
8.5
8
7.5
7
6.5
6
5.5
5
4.5
4
7,000
8,000
9,000
10,000
11,000
12,000
13,000
14,000
15,000
16,000
85 %
80 %
75 %
70 %
65 %
60 %
55 %
50 %
45 %



PART OF 6,800 H. P. OF STIRLING BOILERS OPERATED BY INLAND STEEL COMPANY, INDIANA HARBOR, INDIANA

The heat value per pound of dry coal, as given by the calorimeter, is 15450. Since the moisture in the coal amounted to 0.96%, the heat value per pound of "coal as fired" is $(100-.0096)\times 15450=15302$ B. T. U. The analysis shows that the combustible portion of the coal amounts to $87.76+4.11+4.98=96.85\%$ of the *original* coal, and $15302\div .9685=15800$ nearly. Hence the heat values of the fuel are:

Per pound of coal as fired	15302 B. T. U.	(d)
Per pound of dry coal	15450 B. T. U.	(e)
Per pound of combustible	15800 B. T. U.	(f)

and the efficiencies are:

Based on coal as fired	$10907\div 15302=71.28\%$
Based on dry coal	$11013\div 15450=71.28\%$
Based on combustible	$11477\div 15800=72.64\%$

Efficiency and Combustion-Rate Charts—The charts on pages 188 and 189 illustrate the relation existing between heat value, evaporation, efficiency, heating surface, grate surface and combustion rate, as factors in steam boiler operation, and the two charts may be used separately or jointly, as the conditions of the problem may determine. Only one assumption is made, viz.: that *ten* square feet of heating surface represent *one* boiler horse-power, and that, in consequence, at rating a boiler evaporates 3.45 pounds of water (from and at 212°) per hour per square foot of heating surface. Given the equivalent evaporation and calorific value of the fuel in any case, the efficiency (of the boiler, or of *boiler and grate*, according as the evaporation and heat-value are referred to *combustible* or to *coal as fired*) is shown by the diagonal passing nearest the intersection of the lines corresponding to the other two quantities; in the right-hand chart the corresponding combustion rates, at rating and 50% above rating, are indicated on the diagonal nearest the intersection of the lines for the equivalent evaporation and ratio of heating surface to grate surface.

If, on the other hand, it is desired to obtain a certain rate of evaporation with a boiler of known ratio of heating surface to grate surface, the right-hand chart will indicate the amount of fuel per square foot of grate which must be burned to obtain such evaporation, and by reference to the left-hand chart the heat value of the coal necessary to obtain this evaporation at any given efficiency may be determined.

Distribution of Losses—The efficiency of a boiler, whether based on the combustible or the dry coal, will be found to range from 50% to 80%, and in some cases higher. The difference between the actual efficiency and 100% is the loss occurring in the conversion of the heat energy of the coal into that contained in the steam. This loss is made up of items as follows:

(1) Loss of fuel through the grate.

(2) Unburned fuel carried beyond the bridge wall in the form of soot or small particles.

(3) The heat required to raise the temperature of the moisture in the coal from atmospheric temperature to 212°, to evaporate it at that temperature, and to superheat to the flue-gas temperature the steam thus formed.

(4) The loss due to the presence of hydrogen in the fuel, which forms water which must be evaporated and superheated as in Item 3.

(5) Superheating the moisture in the air supplied, from the prevailing atmospheric temperature to that of the flue-gases.

(6) Heating the products of combustion (excepting the steam) to the flue-gas temperature.

(7) The loss due to incomplete combustion when carbon burns to carbon monoxide (CO) instead of to carbon dioxide (CO_2), and when the volatile gases pass out through the stack unburned.

(8) The loss due to radiation of heat from the boiler and furnace.

It would require an elaborate test to ascertain each one of those items, and in practise it is customary to summarize them as follows:

(a) Loss due to moisture in the coal; this refers to the hydroscopic moisture only. Loss due to moisture formed by burning the hydrogen in the fuel. These two losses in B. T. U.=

$$(9H+W)[212-t+965.8+0.48(T-212)]$$

In which H and W are the proportional part, referred to the combustible, of the

hydrogen and water; t the fire room temperature, and T the breeching temperature.

(b) Loss due to heat carried off in chimney gases, equal in B. T U. to $(T-t)\times 0.24$* $\times$ weight of gases per pound of combustible.

(c) Loss due to incomplete combustion of carbon, forming CO, equal in B. T. U. to

$$C\times\frac{10{,}150\ CO}{(CO_2+CO)}$$

in which CO_2 and CO are percentages by volume of the flue-gases and C is proportional part of carbon in the combustible.

(d) Heat unaccounted for, equal to total heat generated, less the sum of that utilized and the losses (a), (b) and (c). This includes the losses under items (2,) (5) and (8) as above, and loss from unconsumed gases.

A schedule of these losses is called a "Heat Balance." To make it requires an evaporative test of the boiler, an analysis of the flue-gases, an ultimate analysis of the coal, and a calorimeter determination of its heat value.

Example: To illustrate the application of the foregoing the following data from a test of a 517½ H. P. Stirling boiler may be taken:

Steam pressure, absolute	lbs. per sq. in.	175.7
Temperature of stack	deg. F.	481.0
" " fire-room	"	87.
" " feed-water	"	76.7
Weight of coal as fired, per hour,	lbs.	153.1
Moisture in coal	per cent.	2.7
Weight of dry coal, per hour	lbs.	1480.8
Ash and refuse, per hour	lbs.	113.6
Ash and refuse, per hour	per cent.	7.6
Combustible per hour	lbs.	1376.2
Calorific value of combustible per pound	B. T. U.	15696
Analysis of dry coal	C=83.84% by weight.	
	H= 4.72	
	O= 3.77	
	N= 1.65	
	S= 1.07	
	Ash= 4.05	
	100.00%	
Evaporation, actual, per hour	lbs.	14577.9
Moisture in steam	per cent.	.75
Analysis of flue-gases	CO_2=13.5% by volume	
	O= 5.8	
	CO= 0.1	
	N=80.6	
	100.0%	

Since the steam contains .75% moisture, the dry steam per hour amounts to 14,577.9 $\times$ (100.00 − 0.75)=14,468.6 lbs.

The absolute steam pressure being 157.7 lbs. and the temperature of the feed 76.7°, the factor of evaporation is 1.1911; and the equivalent evaporation per hour from and at 212°, is 14,468.6 $\times$ 1.1911=17,233.5 lbs.

17,233.5 ÷ 1376.2 = 12.523 U. E.† per lb. of combustible

17,233.5 ÷ 1489.8=11.568 U. E. per lb. of dry coal.

12.523 $\times$ 965.7 = 12093.4 B. T. U. in steam per lb. of combustible

11.568 $\times$ 965.7‡=11171.2 B. T. U. in steam per lb. of dry coal

The calorific value of the fuel, per pound of combustible, is 15696. The combustible amounts to 100−4.95§=95.05% of the dry coal, whence the calorific value of 1 lb. of the dry coal is 9505 $\times$ 15696 = 14919 B. T. U. and the two efficiencies are,

Efficiency of boiler $\frac{12093.4}{15696}=.7704$

=77.04 per cent. based on the combustible.

Efficiency of boiler and grate= $\frac{11171.2}{14919}$ =

.7487=74.87 per cent., based on the dry coal.

The heat losses are calculated as follows:

(a) Loss due to moisture in coal. The content of moisture referred to combustible is 2.7 ÷ 89.88‖ =.03, and this loss is .03 $\times$ [(212−76.7) +966‡ +0.48 (481−212)]=36.90 B.T.U.

From the ultimate analysis, the hydrogen in the coal is seen to be 4.72%, therefore the loss due to burning hydrogen is,

9 $\times$ .0472 $\times$ [(212−76.7) + 966+0.48 (481 − 212)]=522.5 B. T. U.

(b) To compute the loss of heat in the dry chimney gases per pound of combustible, the weight of the gases must first be ascertained.

From formula [46] page 183 the weight of gases per pound of coal as fired is,

$$\frac{3.032\ N\times C}{CO_2+CO}+(1-A)$$

$$=\frac{3.032\times 80.6\times 0.8384}{13.5+0.1}+(1-0.0495)$$

=16.01 lbs.

Hence the weight of flue-gases per pound of

*Specific heat of chimney gas. †U. E. = Units of evaporation. See p. 72. ‡965.8 would be more exact, but 966 is recommended in the Code of 1885. §Note that this is the real ash as determined from the ultimate analysis, hence is the value to use in determining the B. T U. per pound of dry coal. ‖Combustible consumed per hour ÷ coal as fired per hour. Note the difference between the combustible as a per cent. of the coal as fired, and of the *dry* coal.

combustible is $16.01 \div .8988 = 17.81$ lbs., therefore the loss of heat in stack is

$$17.81 \times 0.24\ (481-87) = 1684 \text{ B. T. U.}$$

(c) Loss due to incomplete combustion. From the ultimate analysis the per cent. of carbon in the combustible is

$$\frac{83.84}{100-4.95} = 88.2\%, \text{ hence this loss is}$$

$$\frac{0.1 \times 0.882 + 10150}{13.5 + 0.1} = 65.35 \text{ B. T. U.}$$

(d) The above determined losses amount to $36.90 + 522.5 + 1684 + 65.35 = 2308.75$ B. T. U. The heat absorbed by the boiler per pound of combustible is 12093.4 B. T. U., hence the total heat accounted for is $12{,}093.4 + 2{,}308.75 = 14{,}402.15$ B. T. U. But the calorific value of one pound of combustible is 15,696 B. T. U., hence

Heat unaccounted for

$$= 15.696 - 14{,}402.15 = 1293.85 \text{ B. T. U.}$$

The heat balance should be arranged thus:

HEAT BALANCE

Total heat of 1 lb. of Combustible = 15696 British Thermal Units.

DISTRIBUTION OF THE HEAT.	B. T. U.	PERCENT
1. Heat absorbed by the boiler . .	12,093.4	77.04
2. Loss due to moisture in coal . .	36.0	.24
3. Loss due to moisture formed by the burning of hydrogen	522.5	3.33
4. Loss due to heat carried away in dry chimney-gases	1,684.0	10.73
5. Loss due to incomplete combustion of carbon	65.35	.42
6. Unaccounted for	1,293.85	8.24
Totals.	15,696.00	100.00

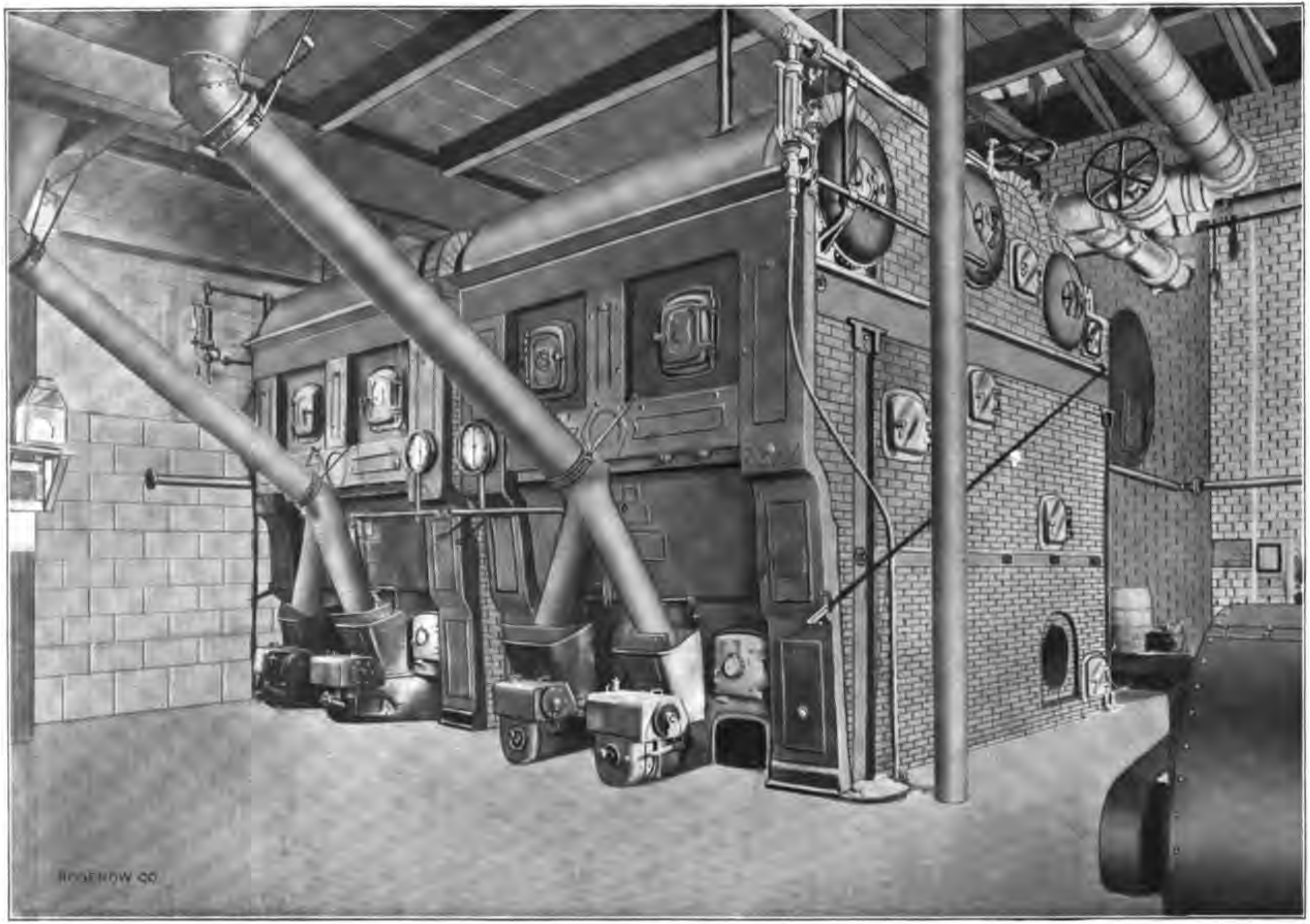
Application of the Heat Balance—Whenever a boiler test supplies data for making a heat balance, it should be made, particularly if the boiler performance is considered unsatisfactory. The distribution of the heat is thus determined and any extraordinary loss can be detected and steps be taken to reduce it.

The heat absorbed to produce steam will range from 50 to 80 per cent., or more, 50% being very poor efficiency and 80% very high; in judging efficiencies the character of fuel must always be considered, since an efficiency that would be regarded as high for one fuel might be very low for another.

The loss due to moisture in the coal is small but appreciable. Therefore coal should be kept under roof, and should not, as in some plants, be wetted before using.

The largest heat loss is due to the chimney-gases. The factors affecting this are the amount of gas and the temperature at which it leaves the boiler. There is a lower limit to the amount of gas, fixed by the minimum air supply with which thorough combustion may be obtained. This limit usually is 18 pounds of air per pound of coal. The limit may be still lower when burning gas or oil. If the air supply is too small, the loss due to carbon burning to carbon monoxide will be increased. The stack temperature is limited, with natural draft, to not much less than 450°, since a lower temperature causes loss of draft and a low heat transfer from gases to water in the boiler. Artificial draft and economizers may reduce these limits, and whether or not they can save enough to compensate for the extra outlay is a problem to be solved for each particular case.

Variation of Efficiency with Rate of Driving—Under any set of conditions there is one rate of evaporation per square foot at which the greatest efficiency will be developed, and there will be a falling off for both higher and lower rates of evaporation. The grade of fuel, skill with which it is fired, the air supply, condition of boiler surfaces, and temperature of steam and of feed water, are some of the factors which affect the result, hence no exact figure for the rate of evaporation which will give the greatest efficiency can be given. Under average conditions it varies from 3 to 4 pounds per square foot of heating surface from and at 212° per hour for water-tube boilers. As the rate is increased the drop in efficiency varies greatly for different types of boiler and the kind of fuel. The matter is of the utmost importance in plants where the boilers must be operated at high rates of evaporations for several hours daily to carry peak loads. The Stirling boiler falls off in efficiency, as the load is increased, much less rapidly than other types, because of the efficient absorption of heat in the rear tube bank where the feed water enters. See "Possibility of Driving at Both High and Low rates of Evaporation," page 27.



BINGHAM COPPER AND GOLD MINING CO., BINGHAM JUNCTION, UTAH, OPERATING 900 H. P. OF STIRLING BOILERS

Horse-Power Rating of Boilers

Work, as the term is used in mechanics, is the overcoming of a resistance through space. The unit of work is the foot-pound.

Power is the *rate* at which work is done, or is the amount of work done in one unit of time. The unit of power in general use among steam engineers is the Horse-power,* which is equivalent to 33,000 foot-pounds per minute, or the work done in lifting 33,000 pounds 1 foot high, or 33 pounds 1,000 feet high, or 1,000 pounds 33 feet high, etc., in one minute.

Horse-Power of Boilers—Boilers for land use are usually rated in "horse-power," and few terms used in engineering are more often misunderstood.

A boiler when in service does not move, hence it does no *work* in the sense in which this word is used in mechanics, therefore it has no *power*. What it really does is to generate steam which acts as a vehicle to convey the energy of the fuel, in the form of heat, to an engine which converts that heat into work and develops power. If every engine developed precisely the same power from an equal amount of heat, the boiler might conveniently be designated as a boiler having the same horse-power as the engine; though inaccurate, the statement could through custom be interpreted to mean that the boiler is of just the capacity required to supply the steam necessary to generate the given horse-power in an engine. Unfortunately, engines of different sizes and types require widely different amounts of steam to produce the same power, hence a boiler which could supply enough steam to produce 500 H. P. in one engine might be able to supply only enough to produce 300 H. P. in another engine which is of less economical design.

Present Meaning of (Stationary) Boiler Horse-Power—To obviate the confusion resulting from an indefinite meaning of the term boiler horse-power, the judges in charge of boiler trials at the Centennial Exposition ascertained that a good engine of the then prevailing types required about 30 lbs. of steam per hour per horse-power developed. In order to establish a relation between the engine power and the size of boiler needed to furnish steam to develop that power, they recommended that an evaporation of 30 pounds of water per hour from an initial feed temperature of 100° F. to steam of 70 pounds gauge pressure be considered as *one boiler horse-power*. The standard thus laid down has been generally accepted by American engineers, and whenever in this country† the term boiler horse-power is used in connection with stationary boilers‡ without special definition, it is to be understood as having the meaning above defined.

To permit easy comparison of results of boiler trials, it is usual to reduce them all to a basis of equivalent evaporation from and at 212° F. One boiler horse-power as above defined is equivalent to an evaporation from and at 212° F. of 34.486 lbs. of water,§ or practically 34.5 lbs., hence,

One boiler horse-power is equal to an evaporation per hour of 30 lbs. of water from 100° F. to steam at 70 pounds pressure; or is equal to an evaporation of 34.5 pounds of water per hour from and at 212° F. It is, therefore, purely a measure of evaporation, and not of power.

Selection of Boilers to Operate an Engine of given Power—To determine the rated horse-power of boiler necessary to develop a given power from an engine, it is necessary to determine the amount of steam re-

*The French horse-power (cheval) is seventy-five kilogrammeters per second $=75\times7.233$ foot-pounds $=$ 542.5 foot-pounds per second, or somewhat less than that used by English-speaking nations, which is equal to 550 foot-pounds per second. Hence,

One horse-power $=1.0139$ cheval
One cheval $=.9864$ horse-power

†In other countries boilers are usually rated, not in horse-powers, but by specifying the quantity of water they are to be capable of evaporating from and at 212°, or under other conditions which can be reduced to equivalent evaporation from and at 212°.

‡When the horse-power of marine boilers is stated it generally refers to and is synonymous with the horse-power developed by the engines which receive steam from the boilers.

§See "Equivalent evaporation from and at 212°," page 69.

TABLE 57

INDICATED HORSE-POWER PER BOILER HORSE-POWER FOR VARIOUS AMOUNTS OF STEAM PER I. H. P. AND FACTORS OF EVAPORATION

Factor of Evaporation.	1.025		1.05		1.075		1.1		1.125	
Actual Water Consumption of Engines per I. H. P. per Hour.	Equivalent Evaporation per I. H. P.	Engine H. P. per Boiler H. P.	Equivalent Evaporation per I. H. P.	Engine H. P. per Boiler H. P.	Equivalent Evaporation per I. H. P.	Engine H. P. per Boiler H. P.	Equivalent Evapor-ation per I. H. P.	Engine H. P. per Boiler H. P.	Equivalent Evapor-ation per I. H. P.	Engine H. P. per Boiler H. P.
10.	10.25	3.36	10.5	3.28	10.75	3.21	11.0	3.13	11.25	3.06
10.5	10.76	3.20	11.02	3.13	11.30	3.05	11.55	2.98	11.81	2.92
11.	11.27	3.06	11.55	2.98	11.85	2.91	12.10	2.85	12.37	2.79
11.5	11.78	2.93	12.07	2.85	12.38	2.78	12.65	2.73	12.94	2.67
12.	12.30	2.80	12.60	2.73	12.90	2.67	13.20	2.61	13.5	2.56
12.5	12.81	2.69	13.12	2.62	13.44	2.56	13.75	2.50	14.06	2.45
13.	13.32	2.59	13.65	2.52	13.98	2.49	14.30	2.41	14.62	2.36
13.25	13.58	2.54	13.91	2.47	14.25	2.42	14.57	2.37	14.91	2.31
13.5	13.84	2.49	14.17	2.43	14.52	2.38	14.85	2.32	15.19	2.27
13.75	14.09	2.45	14.43	2.37	14.72	2.33	15.12	2.28	15.47	2.23
14.	14.35	2.40	14.70	2.33	15.05	2.29	15.4	2.24	15.75	2.19
14.25	14.60	2.36	14.96	2.29	15.32	2.25	15.67	2.20	16.03	2.15
14.5	14.86	2.32	15.22	2.25	15.59	2.21	15.95	2.16	16.31	2.12
14.75	15.12	2.28	15.48	2.21	15.86	2.17	16.22	2.12	16.59	2.08
15.	15.38	2.24	15.75	2.18	16.13	2.13	16.5	2.09	16.87	2.05
15.25	15.63	2.21	16.01	2.15	16.40	2.10	16.77	2.06	17.16	2.01
15.5	15.89	2.17	16.27	2.12	16.67	2.07	17.05	2.03	17.45	1.98
15.75	16.14	2.13	16.53	2.08	16.94	2.04	17.32	1.99	17.72	1.95
16.	16.40	2.10	16.80	2.05	17.20	2.01	17.60	1.96	18.00	1.92
16.25	16.65	2.06	17.06	2.01	17.47	1.98	17.87	1.94	18.28	1.89
16.5	16.91	2.03	17.32	1.98	17.74	1.94	18.15	1.90	18.56	1.86
16.75	17.16	2.00	17.58	1.95	18.01	1.91	18.42	1.87	18.84	1.83
17.	17.42	1.98	17.85	1.93	18.28	1.88	18.7	1.84	19.12	1.80
17.25	17.67	1.95	18.11	1.91	18.55	1.86	18.97	1.82	19.40	1.77
17.5	17.93	1.92	18.37	1.87	18.81	1.83	19.25	1.80	19.69	1.75
17.75	18.19	1.89	18.63	1.84	19.13	1.80	19.52	1.77	19.97	1.72
18.	18.45	1.87	18.90	1.81	19.35	1.77	19.8	1.74	20.25	1.70
18.25	18.70	1.85	19.16	1.78	19.62	1.75	20.07	1.71	20.53	1.67
18.5	18.96	1.82	19.43	1.76	19.89	1.73	20.35	1.69	20.81	1.65
18.75	19.22	1.79	19.69	1.74	20.15	1.71	20.62	1.67	21.09	1.63
19.	19.48	1.76	19.95	1.72	20.42	1.69	20.9	1.65	21.37	1.61
19.25	19.73	1.74	20.21	1.70	20.69	1.67	21.17	1.63	21.65	1.59
19.5	19.99	1.71	20.47	1.67	20.96	1.64	21.45	1.61	21.93	1.57
19.75	20.24	1.69	20.73	1.65	21.23	1.62	21.72	1.59	22.22	1.55
20.	20.50	1.67	21.00	1.63	21.50	1.60	22.0	1.57	22.50	1.53
20.25	20.76	1.65	21.26	1.61	21.77	1.58	22.27	1.54	22.78	1.51
20.5	21.02	1.63	21.52	1.59	22.04	1.56	22.55	1.52	23.06	1.49
20.75	21.27	1.61	21.78	1.57	22.31	1.54	22.82	1.51	23.34	1.47
21.	21.52	1.60	22.05	1.56	22.58	1.53	23.10	1.49	23.62	1.46
21.25	21.77	1.58	22.31	1.54	22.84	1.51	23.37	1.47	23.91	1.44
21.5	22.03	1.56	22.57	1.52	23.11	1.49	23.65	1.45	24.20	1.42
21.75	22.29	1.54	22.83	1.50	23.38	1.47	23.92	1.43	24.47	1.40
22.	22.55	1.52	23.10	1.49	23.65	1.46	24.2	1.42	24.75	1.39
22.5	23.06	1.49	23.62	1.45	24.19	1.42	24.75	1.39	25.31	1.36
23.	23.58	1.46	24.15	1.42	24.73	1.39	25.3	1.36	25.87	1.33
23.5	24.09	1.43	24.67	1.39	25.26	1.36	25.85	1.33	26.43	1.30
24.	24.60	1.40	25.20	1.36	25.80	1.33	26.4	1.30	27.00	1.28
24.5	25.11	1.37	25.72	1.33	26.34	1.30	26.95	1.27	27.56	1.25
25.	25.63	1.34	26.25	1.31	26.88	1.28	27.5	1.25	28.12	1.23
25.5	26.14	1.32	26.77	1.29	27.42	1.26	28.05	1.23	28.68	1.21
26.	26.65	1.29	27.30	1.26	27.95	1.23	28.6	1.20	29.25	1.18
26.5	27.16	1.27	27.82	1.24	28.49	1.21	29.15	1.18	29.80	1.16
27.	27.68	1.24	28.35	1.22	29.03	1.19	29.70	1.16	30.37	1.14
27.5	28.19	1.22	28.87	1.20	29.56	1.17	30.25	1.14	30.43	1.12
28.	28.70	1.20	29.40	1.17	30.10	1.14	30.8	1.11	31.50	1.09
29.	29.72	1.16	30.45	1.13	31.18	1.11	31.9	1.08	32.62	1.05
30.	30.75	1.12	31.50	1.09	32.25	1.08	33.0	1.05	33.75	1.02
31.	31.78	1.08	32.55	1.05	33.32	1.03	34.1	1.02	34.87	.99
32.	32.80	1.05	33.60	1.02	34.40	1.00	35.2	.98	36.0	.96
33.	33.83	1.02	34.65	.99	35.47	.97	36.3	.95	37.12	.93
34.	34.85	.99	35.70	.96	36.55	.94	37.4	.92	38.25	.90
35.	35.87	.96	36.75	.93	37.02	.91	38.5	.89	39.37	.87
36.	36.90	.93	37.80	.90	38.70	.89	39.6	.87	40.5	.85
37.	37.92	.90	38.85	.88	39.77	.86	40.7	.84	41.62	.82
38.	38.95	.88	39.90	.86	40.85	.84	41.8	.82	42.75	.80
39.	39.98	.86	40.95	.84	41.92	.82	42.9	.80	43.87	.79
40.	41.00	.84	42.00	.82	43.00	.80	44.0	.78	45.	.77
41.	42.03	.82	43.05	.80	44.07	.78	45.1	.76	46.12	.75
42.	43.05	.80	44.10	.78	45.15	.76	46.2	.74	47.25	.73
43.	44.08	.78	45.15	.76	46.22	.74	47.3	.72	48.57	.71
44.	45.10	.76	46.20	.74	47.30	.72	48.4	.70	49.5	.69
45.	46.12	.75	47.25	.73	48.37	.71	49.5	.69	50.62	.68

TABLE 57—Continued

INDICATED HORSE-POWER PER BOILER HORSE-POWER FOR VARIOUS AMOUNTS OF STEAM PER I. H. P. AND FACTORS OF EVAPORATION

Factor of Evaporation.	1.15		1.175		1.2		1.225		1.25	
Actual Water Consumption of Engines per I. H. P. per Hour.	Equivalent Evaporation per I. H. P.	Engine H. P. per Boiler H. P.	Equivalent Evaporation per I. H. P.	Engine H. P. per Boiler H. P.	Equivalent Evaporation per I. H. P.	Engine H. P. per Boiler H. P.	Equivalent Evaporation per I. H. P.	Engine H. P. per Boiler H. P.	Equivalent Evaporation per I. H. P.	Engine H. P. per Boiler H. P.
10.	11.5	2.99	11.75	2.93	12.0	2.87	12.25	2.81	12.5	2.76
10.5	12.57	2.85	12.33	2.79	12.6	2.73	12.86	2.69	13.12	2.67
11.	12.65	2.72	12.92	2.67	13.2	2.61	13.47	2.57	13.75	2.58
11.5	13.22	2.60	13.51	2.55	13.8	2.50	14.08	2.45	14.35	2.44
12.	13.8	2.49	14.10	2.44	14.4	2.39	14.70	2.35	15.0	2.30
12.5	14.38	2.40	14.69	2.35	15.0	2.30	15.31	2.25	15.62	2.21
13.	14.95	2.31	15.28	2.25	15.6	2.21	15.92	2.16	16.25	2.12
13.25	15.23	2.26	15.57	2.21	15.9	2.17	16.22	2.12	16.56	2.08
13.5	15.52	2.22	15.86	2.17	16.2	2.13	16.53	2.08	16.87	2.04
13.75	15.81	2.18	16.15	2.13	16.5	2.09	16.84	2.04	17.19	2.01
14.	16.10	2.14	16.45	2.09	16.8	2.05	17.15	2.01	17.5	1.97
14.25	16.38	2.10	16.74	2.05	17.1	2.01	17.45	1.97	17.81	1.93
14.5	16.67	2.06	17.04	2.02	17.4	1.98	17.76	1.94	18.12	1.90
14.75	16.96	2.02	17.33	1.98	17.7	1.94	18.06	1.90	18.44	1.87
15.	17.25	1.99	17.63	1.95	18.	1.91	18.37	1.87	18.75	1.84
15.25	17.53	1.96	17.92	1.92	18.3	1.88	18.68	1.84	19.06	1.81
15.5	17.82	1.93	18.22	1.89	18.6	1.85	18.98	1.81	19.37	1.78
15.75	18.11	1.90	18.51	1.86	18.9	1.82	19.29	1.78	19.66	1.75
16.	18.40	1.87	18.80	1.83	19.2	1.79	19.6	1.76	20.	1.72
16.25	18.68	1.84	19.10	1.80	19.5	1.76	19.9	1.73	20.31	1.69
16.5	18.97	1.81	19.39	1.77	19.8	1.74	20.21	1.70	20.62	1.67
16.75	19.26	1.78	19.68	1.74	20.1	1.71	20.51	1.67	20.93	1.65
17.	19.55	1.76	19.98	1.72	20.4	1.69	20.82	1.65	21.25	1.62
17.25	19.83	1.73	20.27	1.70	20.7	1.66	21.12	1.62	21.56	1.59
17.5	20.12	1.71	20.56	1.68	21.	1.64	21.43	1.60	21.87	1.57
17.75	20.41	1.68	20.85	1.65	21.3	1.62	21.74	1.58	22.18	1.55
18.	20.70	1.66	21.15	1.63	21.6	1.50	22.05	1.56	22.50	1.53
18.25	20.98	1.64	21.45	1.60	21.9	1.57	22.35	1.54	22.81	1.51
18.5	21.27	1.62	21.74	1.58	22.2	1.55	22.66	1.52	23.12	1.49
18.75	21.56	1.60	22.03	1.56	22.5	1.53	22.96	1.50	23.43	1.47
10.	21.85	1.59	22.33	1.55	22.8	1.51	23.27	1.48	23.75	1.45
19.25	22.13	1.56	22.62	1.54	23.1	1.40	23.57	1.46	24.06	1.43
19.5	22.42	1.54	22.91	1.52	23.4	1.47	23.88	1.44	24.37	1.41
19.75	22.71	1.51	23.21	1.49	23.7	1.45	24.19	1.42	24.68	1.39
20.	23.0	1.49	23.50	1.47	24.0	1.43	24.50	1.41	25.00	1.38
20.25	23.28	1.47	23.70	1.45	24.3	1.41	24.81	1.39	25.31	1.36
20.5	23.57	1.45	24.08	1.43	24.6	1.40	25.11	1.37	25.62	1.34
20.75	23.86	1.43	24.37	1.41	24.9	1.39	25.41	1.35	25.93	1.33
21.	24.15	1.42	24.67	1.40	25.2	1.37	25.72	1.34	26.25	1.31
21.25	24.43	1.40	24.96	1.37	25.5	1.35	26.02	1.32	26.56	1.29
21.5	24.72	1.39	25.26	1.36	25.8	1.34	26.33	1.31	26.87	1.28
21.75	25.01	1.38	25.56	1.35	26.1	1.32	26.64	1.30	27.18	1.27
22.	25.3	1.36	25.85	1.33	26.4	1.31	26.95	1.28	27.5	1.26
22.5	25.87	1.33	26.43	1.31	27.	1.28	27.56	1.25	28.12	1.23
23.	26.45	1.30	27.02	1.28	27.6	1.25	28.17	1.22	28.75	1.20
23.5	27.02	1.27	27.61	1.25	28.2	1.22	28.78	1.19	29.37	1.17
24.	27.6	1.25	28.20	1.23	28.8	1.19	29.40	1.17	30.	1.15
24.5	28.17	1.22	28.78	1.20	29.4	1.17	30.01	1.14	30.62	1.12
25.	28.75	1.20	29.37	1.17	30.	1.15	30.62	1.12	31.25	1.10
25.5	29.32	1.18	29.96	1.15	30.6	1.13	31.23	1.10	31.87	1.08
26.	29.9	1.15	30.55	1.12	31.2	1.11	31.85	1.08	32.5	1.06
26.5	30.48	1.13	31.13	1.10	31.8	1.09	32.46	1.06	33.12	1.04
27.	31.05	1.11	31.72	1.09	32.4	1.06	33.07	1.04	33.75	1.02
27.5	31.62	1.09	32.31	1.07	33.	1.04	33.63	1.02	34.37	1.00
28.	32.2	1.06	32.90	1.04	33.6	1.02	34.30	1.00	35.	.98
29.	33.35	1.02	34.07	1.00	34.8	.99	35.52	.97	36.25	.93
30.	34.5	.99	35.25	.97	36.	.96	36.75	.94	37.5	.92
31.	35.65	.96	36.42	.94	37.2	.92	37.97	.91	38.75	.89
32.	36.8	.93	37.60	.92	38.4	.89	39.20	.88	40.	.86
33.	37.95	.90	38.77	.89	39.6	.87	40.42	.85	41.25	.84
34.	39.1	.88	39.95	.87	40.8	.85	41.65	.83	42.5	.81
35.	40.25	.85	41.12	.84	42.	.82	42.87	.80	43.75	.78
36.	41.4	.83	42.3	.81	43.2	.80	44.10	.78	45.	.76
37.	42.55	.80	43.47	.78	44.4	.78	45.32	.76	46.25	.74
38.	43.7	.78	44.65	.76	45.6	.76	46.55	.74	47.5	.72
39.	44.85	.77	45.82	.75	46.8	.74	47.77	.72	48.75	.70
40.	46.	.75	47.00	.73	48.	.72	49.0	.70	50.	.68
41.	47.15	.73	48.18	.71	49.2	.70	50.22	.68	51.2	.66
42.	48.3	.71	49.35	.69	50.4	.68	51.45	.66	52.5	.65
43.	49.45	.69	50.52	.68	51.6	.67	52.67	.65	53.75	.64
44.	50.6	.67	51.70	.66	52.8	.65	53.90	.63	55.	.63
45.	51.75	.66	52.87	.65	54.	.63	55.12	.62	56.25	.62

quired to produce the given power in the engine, then ascertain the size of boiler requisite to generate this steam. The determination of the amount of steam needed by engines of various sizes, types and conditions, can be done only by making actual trials, or by reference to trials on similar engines. sure, engine speed, and point of cut-off. In modern plants using large compound condensing engines with high pressure steam one *boiler horse-power* may be sufficient to develop *two engine horse-power*, including the steam necessary to operate pumps and other auxiliaries. A simple engine of ordi-

TABLE 58

STEAM CONSUMPTION, POUNDS PER INDICATED HORSE POWER*

TYPE OF ENGINE.	Steady Loads.		Variable Loads. 50 to 125 per cent.		Extreme Variations, Railway Work, etc., 0 to 150 per cent.	
	Non-Condensing.	Condensing.	Non-Condensing.	Condensing.	Non-Condensing.	Condensing.
High Speed, simple . .	32	28	34	30	36	31
High Speed, compound .	23	18	25	21	27	22.5
Slow Speed, simple . .	25	21	28	23	31.5	26.5
Slow Speed, compound .	20	15	22.5	18	26	23
High Speed, triple exp. .	17.5	13	20	16	....	....
Slow Speed, triple exp. .	14.5	12.5	17	15	....	....

Table 58 gives a rough approximation of the steam required per indicated horse-power for engines of different types:

Such performance can be expected only from engines of good grade. Plain slide valve engines may use 55 to 60 lbs. per hour per H. P. All similar tables are necessarily very approximate, since the steam consumption will vary with the steam pressure, engine speed, and point of cut-off. nary build, operated non-condensing, will require more than one boiler horse-power per engine horse-power, while direct acting steam pumps will require as much as two or more boiler horse-power per engine horse-power. Consequently, when designing a steam plant it is necessary to determine from the type of engine the steam that will be required, and to make the necessary

TABLE 59

REQUIRED HOURLY EVAPORATION PER BOILER HORSE-POWER AT VARIOUS FEED TEMPERATURES AND STEAM PRESSURES

Feed Temperature Fahr.	STEAM PRESSURE IN POUNDS BY GAUGE.																				
	0	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200
50	29.51	29.29	29.14	29.02	28.92	28.84	28.77	28.70	28.64	28.59	28.54	28.49	28.45	28.41	28.37	28.33	28.29	28.26	28.23	28.20	28.17
60	29.77	29.55	29.40	29.28	29.18	29.09	29.02	28.95	28.89	28.84	28.79	28.74	28.69	28.65	28.61	28.57	28.54	28.51	28.48	28.45	28.42
70	30.04	29.81	29.66	29.54	29.44	29.35	29.27	29.21	29.15	29.09	29.04	28.99	28.94	28.90	28.86	28.82	28.78	28.75	28.72	28.69	28.66
80	30.31	30.08	29.93	29.80	29.70	29.61	29.53	29.46	29.40	29.34	29.29	29.24	29.19	29.15	29.11	29.07	29.03	29.00	28.97	28.94	28.91
90	30.59	30.36	30.20	30.07	29.97	29.88	29.80	29.73	29.67	29.61	29.55	29.50	29.45	29.41	29.37	29.33	29.29	29.25	29.22	29.19	29.16
100	30.88	30.64	30.47	30.34	30.24	30.15	30.07	30.00	29.93	29.87	29.82	29.77	29.72	29.67	29.63	29.59	29.55	29.51	29.48	29.45	29.42
110	31.17	30.93	30.76	30.63	30.52	30.43	30.34	30.27	30.20	30.14	30.09	30.04	29.99	29.94	29.90	29.86	29.82	29.78	29.74	29.71	29.68
120	31.46	31.22	31.05	30.91	30.80	30.71	30.63	30.55	30.48	30.42	30.36	30.31	30.26	30.21	30.17	30.13	30.09	30.05	30.01	29.98	29.95
130	31.76	31.52	31.34	31.20	31.09	30.99	30.91	30.83	30.76	30.70	30.65	30.59	30.54	30.49	30.45	30.41	30.37	30.33	30.29	30.25	30.22
140	32.07	31.82	31.64	31.50	31.38	31.29	31.20	31.12	31.05	30.99	30.93	30.88	30.83	30.78	30.73	30.69	30.65	30.61	30.57	30.53	30.50
150	32.39	32.12	31.94	31.80	31.68	31.58	31.50	31.42	31.35	31.28	31.22	31.17	31.12	31.07	31.02	30.97	30.93	30.89	30.85	30.81	30.78
160	32.71	32.44	32.26	32.11	31.99	31.89	31.80	31.72	31.65	31.58	31.52	31.46	31.41	31.36	31.31	31.27	31.23	31.19	31.15	31.11	31.08
170	33.03	32.76	32.58	32.43	32.31	32.20	32.11	32.03	31.96	31.89	31.83	31.77	31.71	31.66	31.61	31.56	31.52	31.48	31.44	31.40	31.37
180	33.37	33.09	32.90	32.75	32.63	32.52	32.43	32.34	32.27	32.20	32.14	32.08	32.02	31.97	31.92	31.87	31.83	31.79	31.75	31.71	31.67
190	33.71	33.43	33.23	33.08	32.95	32.84	32.75	32.66	32.59	32.52	32.45	32.39	32.33	32.28	32.23	32.18	32.14	32.10	32.06	32.02	31.98
200	34.06	33.77	33.57	33.41	33.28	33.17	33.08	32.99	32.91	32.84	32.77	32.71	32.65	32.60	32.55	32.50	32.45	32.41	32.37	32.33	32.29
212	34.49	34.18	33.98	32.80	33.69	33.58	33.48	33.39	33.31	33.24	33.17	33.11	33.05	33.99	32.94	32.89	32.84	32.80	32.76	32.72	32.68

* See "Economy of Modern Engine Room." *Engineering Magazine*, Oct. 1896.

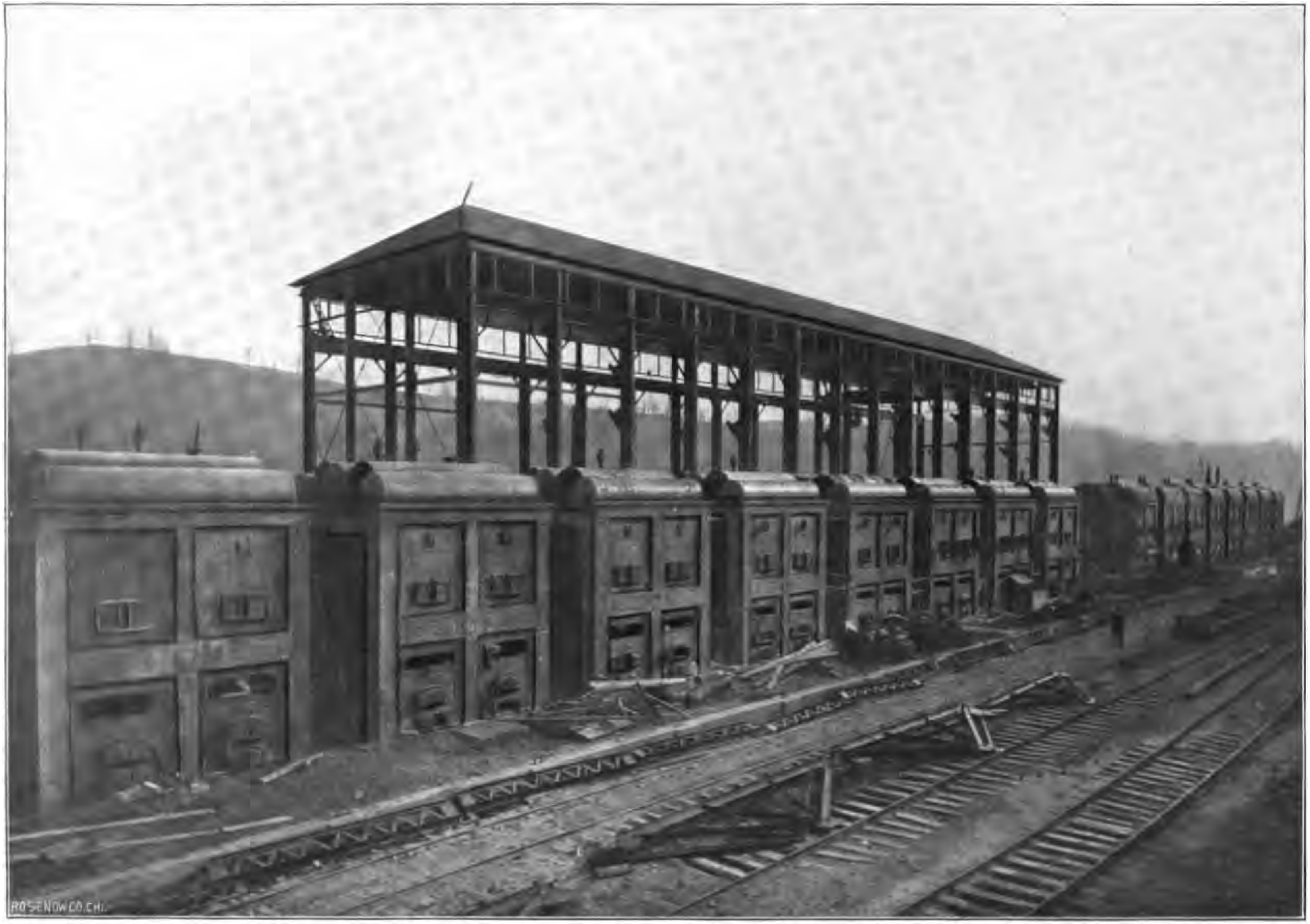
allowance for auxiliaries and boilers undergoing cleaning, and then determine the corresponding boiler capacity.

Number of Units Required—The required boiler horse-power having been determined, the number of units into which it should be divided will depend upon the character of the work to be done. If, for example, there is a day load which is about double the night load, the boiler units should be so proportioned that half the boiler power can be cut out at night. When fixing the number of units, provision should be made for reserve power to allow for repairs, cleaning boilers, and emergencies.

Allowance for Overload—The Committee on Trials of Steam Boilers in their report to the American Society of Mechanical Engineers, said—"A boiler rated at any stated number of horse-powers should be capable of developing that power with easy firing, moderate draft, and ordinary fuel, while exhibiting good economy; and further, the boiler should be capable of developing at least one-third more than its rated power to meet emergencies at times when maximum economy is not the most important object to be attained."

Boiler Horse-Power Tables—When the feed water temperature and the gauge pressure are known, the water per boiler horse-power hour may be taken directly from Table 59. When the actual weight of steam required by an engine per indicated horse-power hour, and the steam pressure and boiler feed temperature are known, the boiler horse-power per engine horse-power can be taken without calculation from Table 57. First, from Table 16 determine the factor of evaporation, then refer to the column under that factor in Table 57 and the tabular value opposite the water consumption of the engine is the boiler horse-power per engine horse-power.

2,500 HORSE-POWER OF STIRLING BOILERS, COTTON STATES AND INTERNATIONAL EXPOSITION, ATLANTA, GEORGIA



UNION STEEL CO., PITTSBURG, PA., OPERATING 25,000 H. P. OF STIRLING BOILERS

Rules for Conducting Boiler Trials

Whenever a boiler test is made it is desirable that the results be recorded in such shape as to permit ready comparison with other tests. In this country boiler tests are usually conducted according to the latest code of rules formulated by a committee of the American Society of Mechanical Engineers, hence this code is here reproduced complete except some portions which touch upon matters more fully treated elsewhere in this book, and to which the reference is given in each case.

RULES FOR CONDUCTING BOILER TRIALS CODE OF 1899.*

I. **Determine at the Outset** the specific object of the proposed trial, whether it be to ascertain the capacity of the boiler, its efficiency as a steam generator, its efficiency and its defects under usual working conditions, the economy of some particular kind of fuel, or the effect of changes of design, proportion, or operation; and prepare for the trial accordingly.

II. **Examine the Boiler,** both outside and inside; ascertain the dimensions of grates, heating surfaces, and all important parts; and make a full record describing the same, and illustrating special features by sketches. The area of heating surface is to be computed from the surfaces of shells, tubes, furnaces, and fire-boxes in contact with the fire or hot gases. The outside diameter of water-tubes and the inside diameter of fire-tubes are to be used in the computation. All surfaces below the mean water level which have water on one side and products of combustion on the other are to be considered as water heating surface, and all surfaces above the mean water level which have steam on one side and products of combustion on the other are to be considered as superheating surface.

III. **Notice the General Condition** of the boiler and its equipment, and record such facts in relation thereto as bear upon the objects in view.

If the object of the trial is to ascertain the maximum economy or capacity of the boiler as a steam generator, the boiler and all its appurtenances should be put in first-class condition. Clean the heating surface inside and outside, remove clinkers from the grates and from the sides of the furnace. Remove all dust, soot, and ashes from the chambers, smoke connections, and flues. Close air leaks in the masonry and poorly fitted cleaning doors. See that the damper will open wide and close tight. Test for air leaks by firing a few shovels of smoky fuel and immediately closing the damper, observing the escape of smoke through the crevices, or by passing the flame of a candle over cracks in the brickwork.

IV. **Determine the Character of the Coal** to be used. For tests of the efficiency or capacity of the boiler for comparison with other boilers, the coal should, if possible, be of some kind which is commercially regarded as a standard. For New England and that portion of the country east of the Allegheny Mountains, good anthracite egg coal, containing not over 10 per cent. of ash, and semi-bituminous Clearfield (Pa.), Cumberland (Md.), and Pocahontas (Va.), are thus regarded. West of the Allegheny Mountains, Pocahontas, (Va.), and New River (W. Va.) semi-bituminous, and Youghiogheny or Pittsburg bituminous coals are recognized as standards.† There is no special grade of coal mined in the Western States which is widely recognized as of superior quality or considered as a standard coal for boiler testing. Big Muddy lump, an Illinois coal mined in Jackson County, Ill., is suggested as being of sufficiently high grade to answer these requirements in districts where it is more conveniently obtainable than the other coals mentioned above.

For tests made to determine the performance of a boiler with a particular kind of coal, such as may be specified in a contract for the sale of a boiler, the coal used should not be higher in ash and in moisture than that specified, since increase in ash and moisture above a stated amount is apt to cause a falling off of both capacity and economy in greater proportion than the proportion of such increase.

V. **Establish the Correctness of all Apparatus** used in the test for weighing and measuring. These are:

1. Scales for weighing coal, ashes, and water.
2. Tanks, or water-meters, for measuring water. Water-meters, as a rule should be used only as a

*From Volume XXI. of the *Transactions* of the American Society of Mechanical Engineers.

†These coals are selected because they are about the only coals which possess the essentials of excellence of quality, adaptability to various kinds of furnaces, grates, boilers, and methods of firing, and wide distribution and general accessibility in the markets.

check on other measurements. For accurate work, the water should be weighed or measured in a tank.

3. Thermometers and pyrometers for taking temperatures of air, steam, feed-water, waste gases, etc.

4. Pressure gauges, draught gauges, etc.

The kind and location of the various pieces of testing apparatus must be left to the judgment of the person conducting the test; always keeping in mind the main object, *that is*, to obtain authentic data.

VI. **See that the boiler is thoroughly heated** to its usual working temperature before the trial. If the boiler is new and of a form provided with a brick setting, it should be in regular use at least a week before the trial, so as to dry and heat the walls. If it has been laid off and become cold, it should be worked before the trial until the walls are well heated.

VII. **The boiler and connections** should be proved to be free from leaks before beginning a test, and all water connections, including blow and extra feed pipes, should be disconnected, stopped with blank flanges, or bled through special openings beyond the valves, except the particular pipe through which water is to be fed to the boiler during the trial. During the test the blow-off and feed pipes should remain exposed to view.

If an injector is used, it should receive steam dirctly through a felted pipe from the boiler being tested.*

If the water is metered after it passes the injector, its temperature should be taken at the point where it leaves the injector. If the quantity is determined before it goes to the injector the temperature should be determined on the suction side of the injector, and if no change of temperature occurs other than that due to the injector, the temperature thus determined is properly that of the feed water. When the temperature changes between the injector and the boiler, as by the use of a heater or by radiation, the temperature at which the water enters and leaves the injector and that at which it enters the boiler should all be taken. In that case the weight to be used is that of the water leaving the injector, computed from the heat units if not directly measured, and the temperature, that of the water entering the boiler.

Let w=weight of water entering the injector.

x = " " steam " " "

h_1=heat units per pound of water entering injector.

h_2=heat units per pound of steam entering injector.

h_3=heat units per pound of water leaving injector.

Then $w+x$=weight of water leaving injector.

$$x=w\frac{h_3-h_1}{h_2-h_3} \qquad [53]$$

See that the steam main is so arranged that water of condensation cannot run back into the boiler.

VIII. **Duration of the Test**—For tests made to ascertain either the maximum economy or the maximum capacity of a boiler, irrespective of the particular class of service for which it is regularly used, the duration should be at least ten hours of continuous running. If the rate of combustion exceeds 25 pounds of coal per square foot of grate surface per hour, it may be stopped when a total of 250 pounds of coal has been burned per square foot of grate.

In cases where the service requires continuous running for the whole 24 hours of the day, with shifts of firemen a number of times during that period, it is well to continue the test for at least 24 hours.

When it is desired to ascertain the performance under the working conditions of practical running, whether the boiler be regularly in use 24 hours a day or only a certain number of hours out of each 24, the fires being banked the balance of the time, the duration should not be less than 24 hours.

IX. **Starting and Stopping a Test**—The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same; the water level the same; the fire upon the grates should be the same in quantity and condition; and the walls, flues, etc., should be of the same temperature. Two methods of obtaining the desired equality of conditions of the fire may be used, viz.; those which were called in the Code of 1885 "the standard method" and "the alternate method," the latter being employed where it is inconvenient to make use of the standard method.†

*In feeding a boiler undergoing test with an injector taking steam from another boiler, or from the main steam pipe from several boilers, the evaporative results may be modified by a difference in the quality of the steam from such source compared with that supplied by the boiler being tested, and in some cases the connection to the injector may act as a drip for the main steam pipe. If it is known that the steam from the main pipe is of the same pressure and quality as that furnished by the boiler undergoing the test, the steam may be taken from such main pipe.

†The Committee concludes that it is best to retain the designations "standard" and "alternate," since they have become widely known and established in the minds of engineers and in the reprints of the Code of 1885. Many engineers prefer the "alternate" to the "standard" method on account of its being less liable to error due to cooling of the boiler at the beginning and end of a test.

X. Standard Method of Starting and Stopping a Test—Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash-pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time and the water level* while the water is in a quiescent state, just before lighting the fire.

At the end of the test remove the whole fire, which has been burned low, clean the grates and ash-pit, and note the water level when the water is in a quiescent state, and record the time of hauling the fire. The water level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and not by operating the pump after the test is completed.

XI. Alternate Method of Starting and Stopping a Test—The boiler being thoroughly heated by a preliminary run, the fires are to be burned low and well cleaned. Note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the water level. Note the time and record it as the starting time. Fresh coal which has been weighed should now be fired. The ash-pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave a bed of coal on the grates of the same depth, and in the same condition, as at the start. When this stage is reached, note the time and record it as the stopping time. The water level and steam pressures should previously be brought as nearly as possible to the same point as at the start. If the water level is not the same as at the start, a correction should be made by computation, and not be operating the pump after the test is completed.

XII. Uniformity of Conditions—In all trials made to ascertain maximum economy or capacity, the conditions should be maintained uniformly constant. Arrangements should be made to dispose of the steam so that the rate of evaporation may be kept the same from beginning to end. This may be accomplished in a single boiler by carrying the steam through a waste-steam pipe, the discharge from which can be regulated as desired. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers, leaving the test boiler to work under a constant rate of production.

Uniformity of conditions should prevail as to the pressure of steam, the height of water, the rate of evaporation, the thickness of fire, the times of firing and quantity of coal fired at one time, and as to the intervals between the times of cleaning the fires.

The method of firing to be carried on in such tests should be dictated by the expert or person in responsible charge of the test, and the method adopted should be adhered to by the fireman throughout the test.

XIII. Keeping the Records—Take note of every event connected with the progress of the trial, however unimportant it may appear. Record the time of every occurrence and the time of taking every weight and every observation.

The coal should be weighed and delivered to the fireman in equal proportions, each sufficient for not more than one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the last of each portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler, and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the test may be divided into several periods if desired, and the degree of uniformity of combustion, evaporation, and economy analyzed for each period. In addition to these records of the coal and the feed water, half hourly observations should be made of the temperature of the feed water, of the flue-gases, of the external air in the boiler room, of the temperature of the furnace when a furnace pyrometer is used, also of the pressure of steam, and of the readings of the instruments for determining the moisture in the steam. A log should be kept on properly prepared blanks containing columns for record of the various observations.

When the "standard method" of starting and stopping the test is used, the hourly rate of combustion and evaporation and the horse-power should be computed from the records taken during the time when the fires are in active condition. This time is somewhat less than the actual time which elapses between the beginning and end of the run. The loss of time due to kindling the fire

*The gauge-glass should not be blown out within an hour before the water level is taken at the beginning and end of test, otherwise an error in the reading of the water level may be caused by a change in the temperature and density of the water in the pipe leading from the bottom of the glass into the boiler.

at the beginning and burning it out at the end makes this course necessary.

XIV. Quality of Steam—The percentage of moisture in steam should be determined by the use of either a throttling or a separating steam calorimeter.* The sampling nozzle should be placed in the vertical steam pipe rising from the boiler. It should be made of ½-inch pipe, and should extend across the diameter of the steam pipe to within half an inch of the opposite side, being closed at the end and perforated with not less than twenty ⅛-inch holes equally distributed along and around its cylindrical surface, but none of these holes should be nearer than ½-inch to the inner side of the steam pipe. The calorimeter and the pipe leading to it should be well covered with felting. Whenever the indications of the throttling or separating calorimeter show that the percentage of moisture is irregular, or occasionally in excess of three per cent., the results should be checked by a steam separator placed in the steam pipe as close to the boiler as convenient, with a calorimeter in the steam pipe just beyond the outlet from the separator. The drip from the separator should be caught and weighed, and the percentage of moisture computed therefrom added to that shown by the calorimeter.

Superheating should be determined by means of a thermometer placed in a mercury-well inserted in the steam pipe. The degree of superheating should be taken as the difference between the reading of the thermometer for superheated steam and the readings of the same thermometer of saturated steam at the same pressure as determined by a special experiment, and not by reference to steam tables.

For calculations relating to, and corrections for, quality of steam, see pages 79 to 85.

XV. Sampling the Coal and Determining Its Moisture—As each barrow load or fresh portion of coal is taken from the coal pile, a representative shovelful is selected from it and placed in a barrel or box in a cool place and kept until the end of the trial. The samples are then mixed and broken into pieces not exceeding one inch in diameter, and reduced by the processes of repeated quartering and crushing until a final sample weighing about five pounds is obtained, and the size of the larger piece is such that they will pass through a sieve with ¼-inch meshes. From this sample two one-quart, air-tight glass preserving jars, or other air-tight vessels which will prevent the escape of moisture from the sample, are to be promptly filled, and these samples are to be kept for subsequent determinations of moisture and of heating value and for chemical analyses. During the process of quartering, when the sample has been reduced to about 100 pounds, a quarter to a half of it may be taken for an approximate determination of moisture. This may be made by placing it in a shallow iron pan, not over three inches deep, carefully weighing it, and setting the pan in the hottest place that can be found on the brickwork of the boiler setting or flues keeping it there for at least 12 hours, and then weighing it. The determination of moisture thus made is believed to be approximately accurate for anthracite and semi-bituminous coals, and also for Pittsburg or Youghiogheny coal; but it cannot be relied upon for coals mined west of Pittsburg, or for other coals containing inherent moisture. For these latter coals it is important that a more accurate method be adopted. The method recommended by the Committee for all accurate tests, whatever the character of the coal, is described as follows:

Take one of the samples contained in the glass jars, and subject it to a thorough air-drying, by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill adjusted so as to produce somewhat coarse grains (less than 1/16-inch), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams, weigh it in a balance which will easily show a variation as small as 1 part in 1,000, and dry it in an air or sand bath at a temperature between 240 and 280 degrees Fahr. for one hour. Weigh it and record the loss, then heat and weigh it again repeatedly, at intervals of an hour or less, until the minimum weight has been reached and the weight begins to increase by oxidation of a portion of the coal. The difference between the original and the minimum weight is taken as the moisture in the air-dried coal. This moisture test should preferably be made on duplicate samples, and the results should agree within 0.3 to 0.4 of one per cent., the mean of the two determinations being taken as the correct result. The sum of the percentage of moisture thus found and the percentage of surface moisture previously determined is the total moisture.

XVI. Treatment of Ashes and Refuse—The ashes and refuse are to be weighed in a dry state. If it is found desirable to show the principal character-

*See pages 79 to 83.

istics of the ash, a sample should be subjected to a proximate analysis and the actual amount of incombustible material determined. For elaborate trials a complete analysis of the ash and refuse should be made.

XVII. Calorific Tests and Analysis of Coal—The quantity of the fuel should be determined either by heat test or by analysis, or by both.

The rational method of determining the total heat of combustion is to burn the sample of coal in an atmosphere of oxygen gas, the coal to be sampled as directed in Article XV of this code.

The chemical analysis of the coal should be made only by an expert chemist. The total heat of combustion computed from the results of the ultimate analysis may be obtained by the use of Dulong's formula, pages 106 and 131.

It is desirable that a proximate analysis should be made, thereby determining the relative proportions of volatile matter and fixed carbon. These proportions furnish an indication of the leading characteristics of the fuel, and serve to fix the class to which it belongs. As an additional indication of the characteristics of the fuel, the specific gravity should be determined.

XVIII. Analysis of Flue-Gases—The analysis of the flue-gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analyses great care should be taken to procure average samples—since the composition is apt to vary at different points of the flue. The composition is also apt to vary from minute to minute, and for this reason the drawings of gas should last a considerable period of time. Where complete determinations are desired, the analyses should be intrusted to an expert chemist. For approximate determinations the Orsat* or the Hempel† apparatus may be used by the engineer.

For the continuous indication of the amount of carbonic acid (CO_2) present in the flue-gases, an instrument may be employed which shows the weight of the sample of gas passing through it.

XIX. Smoke Observations—It is desirable to have a uniform system of determining and recording the quantity of smoke produced where bituminous coal is used. The system commonly employed is to express the degree of smokiness by means of percentages dependent upon the judgment of the observer. The Committee does not place much value upon a percentage method, because it depends so largely upon the personal element, but if this method is used, it is desirable that, so far as possible, a definition be given in explicit terms as to the basis and method employed in arriving at the percentage. The actual measurement of a sample of soot and smoke by some form of meter is to be preferred.

XX. Miscellaneous—In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired certain observations should be made which are in general unnecessary for ordinary tests. These are the measurement of the air supply, the determination of its contained moisture, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

As these determinations are rarely undertaken, it is not deemed advisable to give directions for making them.

XXI. Calculations of Efficiency—Two methods of defining and calculating the efficiency of a boiler are recommended. They are:

1. Efficiency of the boiler

$$=\frac{\text{Heat absorbed per lb. combustible}}{\text{Calorific value of 1 lb. combustible}} \qquad [51]$$

2. Efficiency of the boiler and grate

$$=\frac{\text{Heat absorbed per lb. coal}}{\text{Calorific value of 1 lb. coal}} \qquad [52]$$

The first of these is sometimes called the efficiency based on combustible, and the second efficiency based on coal. The first is recommended as a standard of comparison for all tests, and this is the one which is understood to be referred to when the word "efficiency" alone is used without qualification. The second, however, should be included in a report of a test, together with the first, whenever the object of the test is to determine the efficiency of the boiler and furnace together with the grate (or mechanical stoker), or to compare different furnaces, grates, fuels, or methods of firing.

The heat absorbed per pound of combustible (or per pound of coal) is to be calculated by multiplying the equivalent evaporation from and at 212 degrees per pound combustible (or coal) by 965.7.‡

XXII. The Heat Balance—An approximate "heat balançe," or statement of the distribution of the heating value of the coal among the several items of heat utilized and heat lost may be included in the report of a test when analyses of the fuel and of the chimney-gases have been made. The methods of computing the heat balance and the

*See page 184. †See Hempel's *Methods of Gas Analysis*. (Macmillan & Co.) ‡965.8 is more accurate, and is used throughout this book.

form in which it should be reported, are given in chapter on Steam Boiler Efficiency.

XXIII. Report of the Trial—The data and results should be reported in the manner given in either one of the two following tables,† omitting lines where the tests have not been made as elaborately as provided for in such tables. Additional lines may be added for data relating to the specific object of the test. The extra lines should be classified under the headings provided in the tables, and numbered as per preceding line, with sub-letters *a*, *b*, etc. The Short Form of Report is recommended for commercial tests and as a convenient form of abridging the longer form for publication when saving of space is desirable. For elaborate trials, it is recommended that the full log of the trial be shown graphically, by means of a chart.

DATA AND RESULTS OF EVAPORATIVE TEST

Arranged in accordance with the Complete Form, Code of 1899.

Made by .... of .... boiler at .... to determine ....

Principal conditions governing the trial........

....

Kind of fuel........

Kind of furnace........

State of the weather........

Method of starting and stopping the test ("standard" or "alternate," Art. X and XI, Code)........

1. *Date of trial*........
2. *Duration of trial*........hours.

Dimensions and Proportions.

(A complete description of the boiler, and drawings of the same if of unusual type, should be given on an annexed sheet.)

3. *Grate surface* .... *width* .... *length* ....*area*........ sq. ft.
4. Height of furnace ....ins.
5. Approximate width of air spaces in grate........in.
6. Proportion of air space to whole grate surface .... per cent.
7. *Water-heating surface*........sq. ft.
8. *Superheating surface* .... "
9. Ratio of water-heating surface to grate surface........ —to 1.
10. Ratio of minimum draft area to grate surface........ 1 to ——.

Average Pressures.

11. *Steam pressure by gauge* ....lbs. per sq. in.
12. *Draft between damper and boiler*........ins. of water
13. Force of draft in furnace .... " "
14. Force of draft or blast in ash-pit........ " "

Average Temperatures.

15. Of external air .... deg.
16. Of fireroom .... "
17. Of steam .... "
18. Of feed water entering heater .... "
19. Of feed water entering economizer .... "
20. *Of feed water entering boiler* .... "
21. *Of escaping gases from boiler*........ "
22. Of escaping gases from economizer .... "

Fuel.

23. Size and condition ....
24. Weight of wood used in lighting fire ....lbs.
25. Weight of coal as fired* .... "
26. *Percentage of moisture in coal*‡ .... per cent.
27. *Total weight of dry coal consumed*........ "
28. *Total ash and refuse* ....
29. Quality of ash and refuse........
30. Total combustible consumed........lbs.
31. *Percentage of ash and refuse in dry coal* .... per cent.

Proximate Analysis of Coal.

		Coal.	Combustible.
32.	Fixed carbon	per cent.	per cent.
33.	Volatile matter	"	"
34.	Moisture........	"	——
35.	Ash	"	——
		100 %	100 %
36.	Sulphur, separately determined	"	"

Ultimate Analysis of Dry Coal.
(Art. XVII., Code.)

		Coal.	Combustible.
37.	Carbon (C)	per cent.	per cent.
38.	Hydrogen (H)	"	"
39.	Oxygen (O)	"	"
40.	Nitrogen (N)	"	"
41.	Sulphur (S)	"	"
42.	Ash	"	"
		100 %	100 %
43.	Moisture in sample of coal as received	— "	— "

Analysis of Ash and Refuse.

44. Carbon ....per cent.
45. Earthy matter........ "

Fuel per Hour.

46. *Dry coal consumed per hour* ....lbs.
47. Combustible consumed per hour .... "
48. *Dry coal per sq. ft. of grate surface per hour* "

†To save space, only the table giving the "Complete Form" is here reproduced, but the items printed in italics constitute the "Short Form." *Including equivalent of wood used in lighting the fire, not including unburnt coal withdrawn from furnace at times of cleaning and at end of test. One pound of wood is taken to be equal to 0.4 pounds of coal, or, in case greater accuracy is desired, as having a heat value equivalent to the evaporation of 6 pounds of water from and at 212 degrees per pound. (For more complete information on this point see page 119.) The term "as fired" means in its actual condition, including moisture. ‡This is the total moisture in the coal as found by drying it artificially, as described in Art. XV. of Code.

49. Combustible per square foot of water heating surface per hour ....lbs.

Calorific Value of Fuel.
(Art. XVII., Code.)

50. *Calorific value by oxygen calorimeter, per lb. of dry coal* ....B. T. U.
51. *Calorific value by oxygen calorimeter, per lb. of combustible* .... "
52. Calorific value by analysis, per lb. of dry coal* .... "
53. Calorific value by analysis, per lb. of combustible .... "

Quality of Steam.

54. *Percentage of moisture in steam* ....per cent.
55. Number of degrees of superheating..............deg.
56. Quality of steam (dry steam=unity) ....

Water.

57. *Total weight of water fed to boiler*† ....lbs.
58. Equivalent water fed to boiler from and at 212 degrees .... "
59. *Water actually evaporated, corrected for quality of steam* .... "
60. Factor of evaporation† .... "
61. Equivalent water evaporated into dry steam from and at 212 degrees‡ (Item 59×Item 60.) .... "

Water per Hour.

62. *Water evaporated per hour, corrected for quality of steam* .... "
63. *Equivalent evaporation per hour from and at 212 degrees*‡ .... "
64. *Equivalent evaporation per hour from and at 212 degrees per square foot of water heating surface*‡ .... "

Horse-Power.

65. *Horse-Power developed* (34½ *lbs. of water evaporated per hour into dry steam from and at 212 degrees, equals one horse-power*§ ....H. P.
66. *Builders' rated horse-power* .... "
67. *Percentage of builders' rated horse-power developed*....................per cent.

Economic Results.

68. *Water apparently evaporated under actual conditions per pound of coal as fired.* (*Item* 57÷*Item* 25.) .... lbs.
69. *Equivalent evaporation from and at 212 degrees per pound of coal as fired*‡. (*Item* 61÷*Item* 25.) .... "
70. *Equivalent evaporation from and at 212 degrees per pound of dry coal*‡. (*Item* 61÷*Item* 27.) .... lbs.
71. *Equivalent evaporation from and at 212 degrees per pound of combustible*‡ (*Item* 61÷*Item* 30.) .... "
(If the equivalent evaporation, Items 69, 70 and 71, is not corrected for the quality of steam, the fact should be stated.)

Efficiency.
(Art. XXI, Code.)

72. *Efficiency of boiler; heat absorbed by the boiler per lb. of combustible divided by the heat value of one lb. of combustible*‖ ....per cent.
73. *Efficiency of boiler, including the grate; heat absorbed by the boiler, per lb. of dry coal, divided by the heat value of one lb. of dry coal*.................... .per cent.

Cost of Evaporation.

74. *Cost of coal per ton of*............*lbs. delivered in boiler room* ....$........
75. Cost of fuel for evaporating 1,000 lbs of water under observed conditions ....$........
76. *Cost of fuel used for evaporating* 1,000 *lbs water from and at* 212 *degrees* ....$........

Smoke Observations.

77. Percentage of smoke as observed........per cent.
78. Weight of soot per hour obtained from smoke meter ....ounces.
79. Volume of soot per hour obtained from smoke meter ....cub. in.

Methods of Firing.

80. Kind of firing (spreading, alternate or coking)....................
81. Average thickness of fire....................
82. Average intervals between firing for each furnace during time when fires are in normal condition ....
83. Average interval between times of leveling or breaking up....................

Analyses of the Dry Gases.

84. Carbon dioxide (CO_2) ....per cent.
85. Oxygen (O) .... "
86. Carbon monoxide (CO).................... "
87. Hydrogen and hydrocarbons................ "
88. Nitrogen (by difference) (N).......... "

100 per cent

*See Formula No. 24, page 106. †Corrected for inequality of water level and of steam pressure at beginning and end of test. †Factor of evaporation, see page 70. ‡The symbol "U. E." meaning "Units of Evaporation," may be conveniently substituted for the expression "Equivalent water evaporated into dry steam from and at 212 degrees," its definition being given in a foot-note. See page 72. §Held to be the equivalent of 30 lbs. of water per hour evaporated from 100 degrees Fahr. into dry steam at 70 lbs. gauge pressure. See page 195. ‖In all cases where the word combustible is used, it means the coal without moisture and ash but including all other constituents. It is the same as what is called in Europe "coal dry and free from ash." See foot-note on page 112.

TABLE 60

TESTS ON STIRLING WATER-TUBE BOILERS BURNING VARIOUS GRADES OF FUEL

Name and Place	Testing Expert	Duration of Test	Boiler Horse-Power			Temperatures		Moisture in Steam Per Cent.	Draft in Ins. of Water	Coal Burned per hr. per sq. ft. of Grate Lbs.	Water Evaporated from and at 212° per Pound of		Kind of Fuel
			Rated	Per Cent. Developed.									
		Hours		Above Rating	Below Rati'g	Feed Water Fahr.	Flue-Gases. Fahr.				Dry Coal	Combustible	
Mergenthaler Linotype Co., Brooklyn, N. Y.	J. E. Denton	9.47	300	6.2	....	45.4	406.7	.81	.55	14.90	9.70	11.55	Anthracite Pea.
Mergenthaler Linotype Co. Brooklyn N. Y.	" "	6.00	100	50.5	....	50.0	621.0	.55	.60	24.18	8.48	9.49	" "
Mergenthaler Linotype Co., Brooklyn, N. Y.	" "	7.00	200	51.0	....	55.5	554.7	.61	.60	25.77	8.31	9.57	" "
Midvale Colliery, Wilburton, Pa . . .	F. M. Faber	6.83	200	8.5	....	64.0	482.0	.30	.30	8.28	8.22	10.87	Anthracite Buckwheat.
Lehigh & Wilkes-Barre Coal Co. . .	G. H. Barrus	9.24	125	28.0	....	42.0	485.0	Dry	.16	14.84	10.07	11.45	No. 2 Buckwheat.
Lehigh & Wilkes-Barre Coal Co. . .	" "	7.88	125	132.0	....	41.0	654.0	.50	.16	28.28	9.31	10.65	" "
Blackstone Mfg. Co., Blackstone, Mass .	Barrus and Evans	10.49	250	.....	2.0	54.0	436.0	(f)	.21	13.46	10.91	12.05	Cumberland.
Toledo Water Works., Toledo, O. . .	M. E. Cooley	16.33	300	6.0	....	116.0	503.0	.41	.45	21.7	10.38	11.79	"
Portland (Me.) Street Railway Co. . .	G. H. Barrus	8.57	250	.....	15.0	210.0	487.0	.06	.11	11.69	11.02	12.06	"
Public Works., Bangor, Me.	Thos. Pray, Jr.	10.00	350	7.5	....	186.6	482.5	.56	.29	17.01	11.10(a)	11.75	Big Vein, Cumberland.
Old Colony S reet Ry. Co., Taunton, Mass	M. Mellinger	14.00	650	.....	11.0	147.0	572.0	.60	.50	16.17	11.9	12.93	George's Creek.
Windber Electric Co., Windber, Pa. . .	M. Mellinger	10.00	250	3.0	....	55.3	383.2	1.00	....	15.3	12.06	12.52	Clearfield.
West Chicago Street Railway Co. . . .	Hunt and Jones	18.00	800	19.3	....	82.9	552.0	.09	.50	18.71	11.20	11.63	New River.
Waltham (Mass.) Bleachery and Dye Works.	Dean & Main	10.00	1125	4.8	....	128.4	497.0	.58	.186	13.3	12.44	13.03	" "
General Electric Co., Schenectady, N. Y .	Denton, Barrus	10.00	517.5	.....	3.5	76.7	481.0	1.02	(b)	(b)	11.568	12.523	" "
General Electric Co., Schenectady, N. Y.	Dean & Main	10.58	517.5	36.2	....	71.2	550.0	1.01	(b)	(b)	10.833	11.579	" "
Belle City Street Ry. Co., Racine, Wis .	D. P. Jones	8.00	200	15.0	....	168.0	480.0	.91	.28	13.32	11.44	12.1	Youghiogheny.
Pittsburg & West End Passenger Ry. Co.	Fitzgerald & Faber	10.00	300	19.6	....	39.0	494.0	.25	.05	20.2	10.34	11.44	" Slack, 2nd Pool.
Pittsburg & West End Passenger Ry. Co.	Fitzgerald & Faber	6.00	300	76.0	....	40.0	603.0	.32	.625	33.9	8.99	10.31	" " " "
American Steel Hoop Co., Youngstown, O.	F. M. Faber	7.00	250	32.8	....	141.0	493.0	.075	.45	14.6	10.18	11.29	" 2nd Pool.
Sherwin Williams Co., Cleveland, O. . .	Hayford Moore	8.00	300	4.41	....	169.0	535.0	.....	(b)	(b)	11.58	12.86	Pittsburg Run of Mine.
B. F. Goodrich Co., Akron, O. . . .	P. Hauser	8.00	300	22.6	...	70.9	457.0	.....	.35	17.63	9.26	11.26	Bituminous Slack.
Frasch Soda Co., Cleveland, O. . . .	Chas. Hyde	6.00	200	16.6	....	80.0	590.0	.....	.41	19.6	8.93	11.12	" "
Marshall Kennedy Milling Co., Allegheny Pa.	D. Ashworth	8.00	600	.....	18.0	71.0	421.0	1.00	.54	17.2	9.67	11.1	" "
Oxford Paper Co., Rumford Mills, Me. .	M. Mellinger	9.25	500	.....	0.1	34.0	584.0	.80	(b)	(b)	7.78	8.69	Nova Scotia Culm.
Newark Traction Co., Newark, O. . .	C. H. Delaney	8.00	304	5.6	....	116.0	417.0	.65	.35	.....	0.895(c)	......	Natural Gas.
Los Angeles Electric Co. Los Angeles Cal.	H. M. Boon	5.00	500	28.28	....	123.0	547.5	.54	....	.....	16.335(d)	......	California Crude Oil.
Pacific Light & Power Co., Los Angeles Cal.	Cory and Boon	10.00	335	1.79	....	63.65	454.0	.56	.14	.....	15.31 (d)	......	" " "
Pacific Light & Power Co., Los Angeles, Cal.	Cory and Boon	10.00	335	97.36	....	62.05	720.0	.69	.34	.....	13.85 (d)	......	" " "
Jung Brewing Co., Cincinnati, O. . . .	F. M. Faber	10.00	250	.....	4.8	69.0	600.0	1.23	.35	17.8	10.24	11.21	Kanawha, W. Va., Screenings.
Lundell Street Ry. Co., St. Louis, Mo. .	W. H. Bryan	9.00	300	15.0	....	47.0	448.0	.14	.77	.....	7.82	8.92	Illinois Lump.
Belle City Street Ry. Co., Racine, Wis .	D. P. Jones	8.00	200	14.0	....	155.0	485.0	.98	.38	17.81	8.59	9.34	" "
West Chicago Street Ry. Co., Chicago, Ills	Hunt and Jones	18.00	800	.....	11.5	70.1	548.7	.96	.52	20.57	.756	9.01	Streator, Ills., Run of Mine.

(a) Per pound of wet coal. (b) American stokers and forced draft. (c) Per cubic foot of gas. (d) Per pound of oil. (f) Superheated.

Boilers for Mining Service

To prove satisfactory for mining service a boiler must be capable of meeting a wider range of requirements than is ordinarily met with in other industries. It must be simple in construction so that repairs can be quickly made with the limited equipment usually available; it must be safe in operation because skilled boiler attendants are often unavailable; it must be easy to force in emergencies, and its design of fire-box must be such as to permit use of wood, oil, and high or low grade coal. The boiler must be reasonable in first cost, because of the uncertainty as to the life of many mines, and it must also be easy to transport into places difficult of access. An especially important requirement is that the boiler can be opened, cleaned and closed in the shortest possible time, not only because of the expense of keeping it out of commission, but because of the high cost of labor. The single item of labor-cost of cleaning some types of boilers has prevented their extensive use for mining plants.

A careful reading of the preceding description of the Stirling boiler will demonstrate that it perfectly meets each of these requirements, and surpasses any competing type in its adaptation to varying fuel requirements, and ease and cheapness of cleaning. A more convincing evidence that it is justly regarded as the "Ideal Boiler for Mine Use" is found in the fact that over 200,000 horsepower of Stirling boilers are now in successful operation in mine and smelter plants.

Boilers supplying Hoisting Engines are often blamed for insufficient capacity, wet steam, etc., when the fault is due not to the boiler, but to improper piping. The hoist is usually some distance from the boiler, and often the steam pipe is so imperfectly covered that between lifts large quantities of steam condense and the water thus formed is swept into the cylinder when the engine starts. When the throttle is thrown open the *momentary draft* on the boiler may be many times greater than its rated capacity and in such cases, irrespective of the kind of boiler, there may be a momentary lift of water. To prevent this it is common practise to provide the boiler with an auxiliary steam drum, but this is a makeshift and not a cure, as will now be shown.

Condensation in the pipe line cannot be prevented, but it can be largely reduced, and the evident means are an efficient covering of ample thickness, and a reduction of the capacity. A reliable method of extracting all water of condensation before the steam enters the engine is necessary.

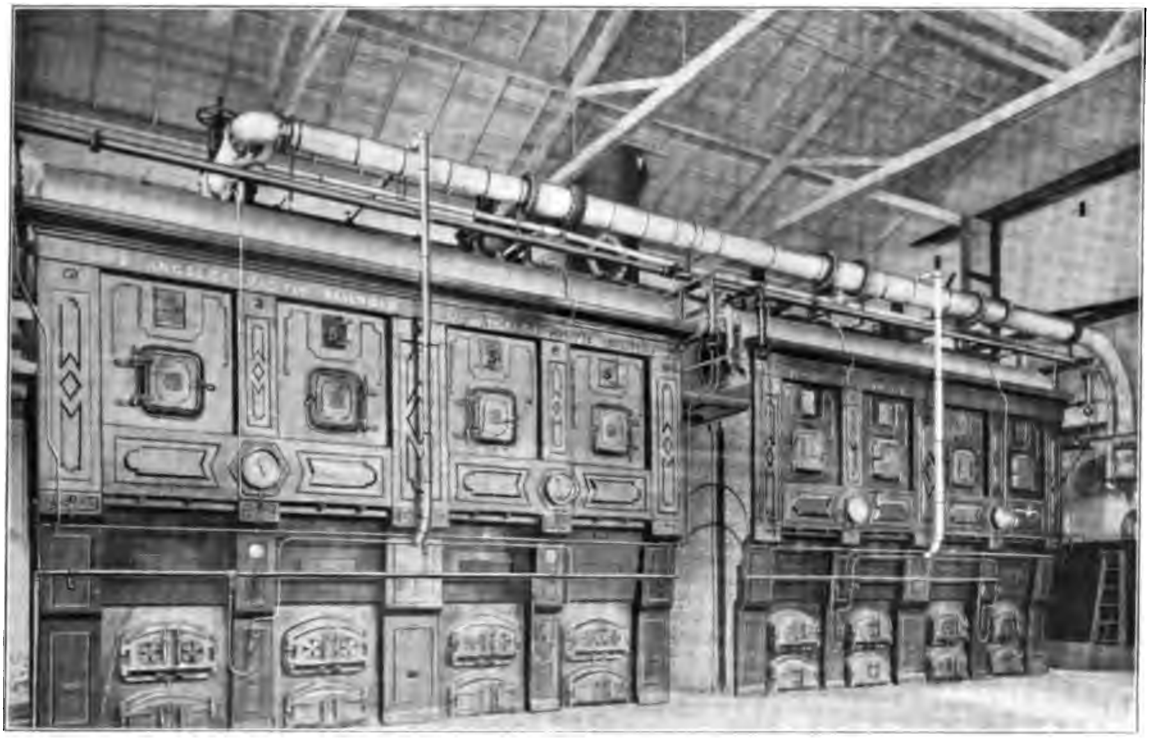
Consider the case of a tank supplied with water through a pipe connected to a distant reservoir. It is evident that if one wants to fill the tank say every minute, yet allow only one-quarter of a minute to do the actual filling, and shut off the pipe the other three-quarters of a minute, then the column of water in the pipe is not only started and stopped at every filling, but the capacity of the pipe will have to be four times as great as would be the case if the water ran all the time into a storage tank from which the other tank could be filled at the same intervals as before. Provided the original reservoir feeding the pipe were of sufficient size to supply the requisite volume of water, any further increase in its size could not alter in the slightest degree the action as above described.

If in the above case a boiler be substituted for the reservoir, and a hoisting engine for the tank to be periodically filled, then the storage tank from which it is to be filled corresponds exactly to the steam drum, hence it is evident that when the drum is placed *on the boiler*, it is at the wrong end of the line. Its proper place is as near to the engine as it can possibly be put, and its cubical capacity should be not less than one-seventh the *actual* volume of steam used per minute by the engine. It should be well covered, and provided with an absolutely reliable drainage apparatus, and a gauge glass to indicate when that apparatus fails to work. Steam traps are not always reliable, and an automatic drainage pump is better.

The area of the steam pipe may next be figured. The maximum number of strokes

OLD DOMINION COPPER MINING & SMELTING CO., GLOBE, ARIZONA, OPERATING 2,800 H. P. OF STIRLING BOILERS

per minute of the engine being known, disregard the cut-off and assume the cylinder is entirely filled with steam at each stroke, then compute the diameter of steam pipe between engine and drum based on steam velocity of 6,000 feet per minute. This velocity will actually exist *during the part of the stroke up to cut-off*, since the velocity of the steam at that time will be the same as it would be were the engine taking steam during the entire stroke.



LOS ANGELES-PACIFIC RAILROAD, LOS ANGELES, CAL. OPERATING 1,300 H. P. OF STIRLING BOILERS

To reduce the condensation, the pipe between boiler and drum should be no larger than actually required, and it will be entirely safe to figure its area in exactly the same manner as above shown for engine supply pipe, except that the steam velocity may be assumed as 8,000 or even 9,000 feet per minute. The actual velocity through the pipe thus determined will be less than these figures, because the volume of steam assumed as basis of the calculation is that which would be required to fill the entire engine cylinder if there were no cut-off; actually the flow in the engine supply pipe will continue only up to time the engine cuts off; the steam being elastic and the drum acting as a receiver and pressure steadier (like the rubber bag in a gas engine supply pipe) the flow of steam between boiler and drum will be practically continuous, hence its velocity will be less than assumed in the calculation.

The decrease in pipe size will reduce the surface available for condensation. The storage drum next to the engine is preferably made vertical, and to assist in separating any water a vertical partition should extend from the top to within 18 inches of the bottom, and both steam inlet and outlet should be near the top. To secure the maximum of volume with minimum of exposed surface, the diameter and length should not differ greatly.

Regardless of the purpose for which a steam plant may be used, there are many more cases than commonly supposed where an arrangement of drum and piping as above described would be a profitable investment.

PART OF 11,600 H. P. OF STIRLING BOILERS, WASHOE SMELTER, ANACONDA COPPER MINING CO., ANACONDA, MONTANA

Principles of Steam Piping

In the design of a steam plant no detail merits more careful consideration than the steam piping. Not only is it frequently overlooked, but the evils resulting from defective design are usually attributed to other parts of the equipment.

The nature of the material to be conveyed by the pipe must be considered, as the requirements for steam are entirely different from those of water, oil, or gas. The principles governing steam pipe design are: (1) The moment steam leaves the boiler it loses heat and some of it must condense. (2) Water of condensation is an evil, and since its formation cannot be wholly prevented a perfect pipe system must provide means of removing it as fast as it forms. (3) There can be no flow of steam without a corresponding drop of pressure. (4) Drop of pressure of steam does not cause a loss of energy. (5) The mechanical design must provide ample strength, provision for expansion, and valves of suitable type properly located.

No perfect heat insulator is known; the loss of heat from steam pipes by radiation may be reduced by methods given in the next chapter, but it cannot be wholly prevented, hence some water of condensation must form. If this water as fast as it is formed is carried to the engine it will cause troubles which will later be pointed out. If the pipe contains low spots or "pockets" where the water can accumulate it will gradually decrease the effective pipe area until the steam velocity is increased to a sufficient degree to lift the water and sweep it along the pipe. This is especially liable to happen when the demand for steam is irregular; when the flow is small the water will settle into the pockets but when the heavy load is suddenly thrown on, the resulting rush of steam will carry the water

TABLE 61

STANDARD DIMENSIONS OF WROUGHT IRON AND STEEL STEAM, GAS AND WATER PIPE*

Diameter.			Nominal Thickness.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Length of Pipe Containing one Cubic Foot.	Nominal Weight per Foot.	Number of Threads per Inch of Screw.
Nominal Internal.	Actual External Diameter	Approximate Internal Diameter		External.	Internal.	External.	Internal.	Metal.	External Surface.	Internal Surface.			
Inch.	Inches.	Inches.	Inches.	Inches.	Inches.	Sq. Inch.	Sq. Inch.	Sq. Inch.	Feet.	Feet.	Feet.	Pounds.	
1/8	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513.	.241	27
1/4	.54	.364	.088	1.696	1.144	.229	.1041	.1240	7.075	10.49	1383.3	.42	18
3/8	.675	.494	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.559	18
1/2	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14
3/4	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270.	1.115	14
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.668	11½
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11½
1½	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.678	11½
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11½
2½	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.739	8
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8
3½	4.	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8
4½	5.	4.508	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.49	8
5	5.563	5.045	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.762	8
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	23.271	8
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.177	8
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.065	8
11	11.75	11.	....	36.914	34.558	108.434	95.033	13.401	.325	.347	1.51	45.028	8
12	12.75	12.	....	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	48.985	8

*From Crane Company's Catalog.

bodily with it. Since water is practically incompressible its effect when traveling at high velocity differs little from that of a solid body of equal weight, hence its impact against elbows, valves, or other obstructions, is equivalent to a heavy hammer blow, and frequently the pipe is ruptured. If the quantity of water is insufficient to produce such serious results, it will certainly cause knocking and vibrations in the pipes, the final consequence of which will be leaky joints. When the water reaches the engine its effects will vary from disagreeable knocking to destruction of the engine. In such cases the usual procedure is to blame the boiler for producing "wet" steam, but the fallacy of this view can easily be shown. Assume a compound condensing engine which develops 200 H. P. under 125 lbs. gauge pressure, and uses 18 lbs. of steam per horse-power hour. Assume that the air temperature is 80°, and that the steam pipe is 6 inches dia., and 150 feet long. The engine will use 3,600 lbs. of steam per hour. and under the given conditions the steam pipe, if uncovered, will condense about 225 lbs. per hour, or 6.25% of the total. If by proper covering 80% of this condensation can be prevented, the remainder will be 1.25% of the total, and the boiler is in no sense responsible for this amount of moisture in the steam. In many cases where properly designed boilers are blamed for wet steam, a similar analysis would locate the trouble in the steam pipes.

Removal of the Water—It follows that efficient means of removing the water of condensation are absolutely imperative. The sketches in Fig. 43 will illustrate the principles governing this removal. Each sketch represents an elevation of a system of steam piping.

M indicates the boiler and *E* the engines. Sketch *A* shows the simplest scheme of piping. All the water of condensation will flow into the engines, unless removed by separators at *S*. If all the engines are shut down, water will collect in the pipes, and unless it be drained off by "bleeders" it will cause trouble when an engine is started.

In sketch *B* the engine connections are taken from the *top* of the main, hence most of the water will flow to the "dead end" or drop leg *X*; the small amount carried over to the engine may be removed by the separators *S*. The water which collects at *X* must be continuously removed by a trap or pump, and if this be done the system will be drained, and the piping maintained at an even temperature, whether the engines are operating or not.

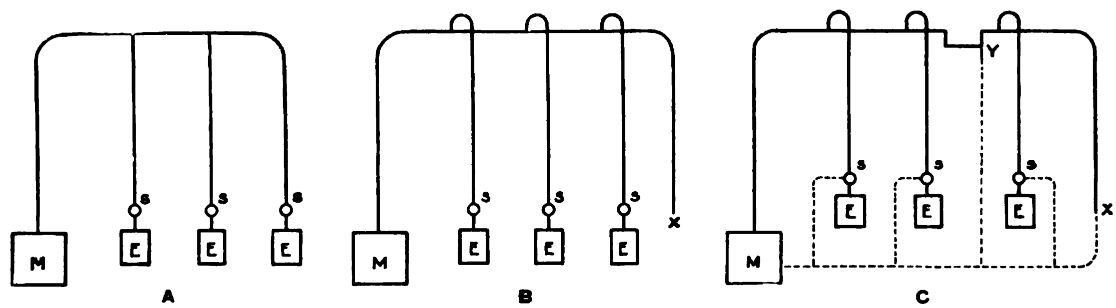


FIG. 43. LINE DIAGRAMS SHOWING ELEVATION OF STEAM PIPE SYSTEMS

If the trap or pump at *X* be replaced by a drain pipe which is connected below the boiler water-line, as in sketch *C*, the steam main is at once converted into a ***high pressure gravity steam-heating system***, and it will automatically drain itself, provided the drop in pressure in the main is not sufficient to maintain a level of water in the drain higher than the point *X*. If the level due to the drop be lower than the separators *S*, then their drips also could be connected to the drainage pipe, and in that case the engine supply pipes will form part of the high pressure heating system, and will be self draining. Should it be necessary to make a dip or pocket in the steam main, as at *Y*, its lowest point

should also be connected to the drainage system. Thus arranged, the entire system will maintain its circulation, there will be no straining due to heating and cooling, and the system will be self-draining whether the engines be working or not. All such drainage connections should be provided with check valves, as shown in Fig 44.

When boilers are situated a sufficient distance below the engines (as in some mills), the installation of piping arranged as a gravity-return system is entirely feasible and the results leave nothing to be desired. In most cases the necessary difference of level cannot be obtained, and a mechanical equivalent for it must be installed. The usual method is to install either steam traps, which will deliver the water of condensation into the heater or hot well; or steam pumps, which will return the water directly into the boiler at practically steam temperature. A number of drain pipes are connected to each trap or pump. Traps are not as reliable as pumps, hence the latter should be preferred if the amount of work to be done will justify their greater cost.

Branches from the mains should *never* be taken from the bottom. When possible they should be taken from the top, and a horizontal partition in the center of the tee will still further improve the action, since in case of a sudden demand for steam at the engine, the water flowing along the bottom of the main cannot be lifted into the outlet pipe.

The pitch of all pipe should be in the direction of the steam travel. Whenever a rise in the main is necessary, a drain, as at *Y* in sketch *C*, should be tapped into the lowest point just below the rise. The mains and all important branches should terminate in a drop-leg, as at *X*, and every such drop-leg or other low spot in the system, should be connected to the drainage pump. A similar connection should be made to every fitting which is of such shape that it can form a water pocket. An effective method of draining the mains is to run at a suitable distance below them a small drainage pipe, about 1¼ to 1½-inch dia., which is connected to all low spots in the piping or fittings, and conveys the water to the drainage pump. To prevent this pipe from delivering steam and water into any section of the main from which steam may have been cut off by the regular valve systems, a swinging check valve should be inserted into each connection between the main and the drainage pipe. This valve can be placed at an angle of nearly 45°, as in Fig. 44, so that the check valve disk is nearly vertical, hence requires practically no head of water to open it when in action.

Each engine supply pipe should have its own separator placed as near the throttle as possible, and the drains from these separators can be connected to a drainage system also.

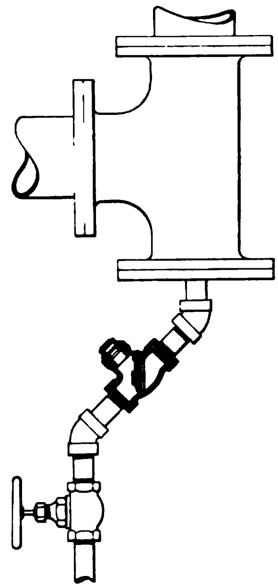


FIG. 44. LOCATION OF CHECK VALVE IN DRAIN PIPE

Drain pipes which are occasionally opened by hand may be useful for blowing out large quantities of water at intervals, but their action must be intermittent, and they are usually neglected.

It is questionable whether the small additional cost of providing a steam plant with an efficient drainage system as above described could be invested to better advantage. It will insure a positive saving by reducing the initial condensation in the engine; it will insure better cylinder lubrication with a reduced supply of oil; and it will return the water of condensation to the boilers at practically steam temperature. It will eliminate the straining of pipes due to cooling when engines are shut down, hence

obviate leaks; and it will remove all danger of wrecking engines by water.

Size of Steam Pipes—The larger the pipe, the greater the surface, hence the greater the amount of condensation. The usual practise is to limit the steam velocity in mains to 6,000 feet per minute, yet there are many cases where this figure could be increased with advantage. That this is not more frequently done is due to the impression that drop of steam pressure causes a loss of energy. In the explanation of the throttling calorimeter, page 79, it was shown that there is no loss, because the difference in energy between the steam at the higher and lower pressures is converted into heat which evaporates moisture and superheats the steam. But if the pressure drop is increased the steam velocity is also increased, hence the pipe area is decreased; but in that case the exposed surface is also decreased, hence the amount of condensation is proportionately decreased. Therefore, by increasing the drop in pressure, the condensation is not only decreased, but the heat liberated by the drop will evaporate all or part of the water which is formed, hence the steam which reaches the engine will be drier than if it had been delivered through a larger pipe; consequently the drop in pressure causes an actual saving in heat instead of a loss. To deliver the steam at the engine at a given pressure it is necessary only to increase the boiler pressure by the amount of the drop, and if this be done it is evident that the size of steam mains and the resulting condensation will both be decreased, with consequent improvement in the action of the engine. When steam is to be conveyed through long lines of piping the advantage of raising the boiler pressure and keeping the mains small will be very marked. The entire method is exactly parallel to standard practise in electrical distributing systems, where the generator voltage is adjusted to suit the loss in the feeder lines; this loss corresponds to the drop in pressure in the steam pipe, and by reason of the drop both feeder wire and the steam pipe can be of reduced size. At the electrical receiving station a storage battery is often installed, and its analogue in the steam plant is a receiving drum placed near the engine. The effect of this drum is to cause a nearly constant flow of steam through the pipe which connects it to the boiler or main header, while the engine draws out steam intermittently. The action is more fully described under caption "Boilers Supplying Hoisting Engines," page 209, and is worthy of careful study. To carry the analogy still farther, it is known that in an electrical distribution system covering a wide area, storage batteries installed at particular points absorb energy when there is a surplus, and liberate it when there is a deficiency, and thus steady the voltage on the entire system, and obviate necessity of supply mains of excessive size. In precisely the same way a steam pipe system can be designed so that storage drums at ends of long lines, etc., will receive steam during the time the engine cut-off is in action, and thereby steady the pressure, and enable the pipe sizes to be so reduced that the area of exposed surface saved in the pipe exceeds that added by the drum, hence not only will the pressure at the engine be kept more steady and the vibrations of the pipe be reduced, but the condensation will be decreased, and the removal of the water which is formed will be more easily accomplished. There are many cases in all kinds of plants where a practical application of these principles would greatly improve the operation of the machinery.

The Constructive Details of steam piping have been so well worked out that only the leading points need be touched upon. The defects usually noted are flanges which are too light, and inadequate provision for expansion. The use of pipe bends of long radii, instead of cast elbows, is becoming more general, and they can usually be so placed as to take up all the expansion without the use of slip-joints with stuffing boxes. The latter almost invariably cause trouble; if their use is compulsory the pipe must be so anchored that the slip-joint cannot pull itself apart.

Duplicate Pipe Systems are now seldom installed. They increase first cost, multiply the number of valves and joints, increase the condensation, and are of questionable utility. It will generally be found that the same or less money, if invested in a single pipe system *properly designed and built*, will

insure equally good service at a lower cost for maintenance.

Valves should be so located that they cannot form water pockets when either opened or closed. Globe valves cause a drop of pressure, but as explained, this does not cause a loss of energy, but a conversion of it into heat; the globe forms a water pocket unless it is set with its stem horizontal, while a gate valve may be set with spindle vertical or at an angle as occasion demands. Valves over 5 to 6 inches diameter should be provided with by-pass, to enable them to be easily opened, and to permit steam to be admitted very slowly into the pipe which can thereby be gradually warmed up, and prevent water hammer.

Boiler Valves—The feed valve should *always* be a globe. A gate valve cannot be closely regulated, and often clatters owing to the pulsations of the feed pump.

Boiler stop valves should be so placed that water cannot collect above them. Thus, if the pipe rises for a distance above the boiler nozzle before turning horizontal, the stop valve should be in the horizontal run. When a long bend leads out of the boiler nozzle the stop valve should be at the highest point of the bend. When it is impossible to avoid locating the valve so that water can accumulate above it when closed, a drain pipe should be provided. The best practise is to provide two valves, one placed as near the boiler as practicable, and the other at the junction of the boiler pipe and the main header, with a drain pipe placed between the two valves to remove any water due to leakage through the header valve.

Automatic Stop Valves are coming more into use, and in some European countries their installation, when several boilers are operated together, is prescribed by law. When several boilers feed into the same header it is evident that if a tube ruptures the steam from the main will rush toward the disabled boiler, hence all the boilers will tend to discharge through the one which is disabled. The sudden rush of steam thus caused will lift water, which may be swept along to the engine and wreck it. The difficulty of closing the stop valve of the disabled boiler is evident. This can be obviated by selecting for position nearest the boiler an *automatic stop valve*, which will close when the pressure in the main slightly exceeds that in the boiler, and open when the boiler pressure rises again. Such valves cost but little more than ordinary stop valves, and should a tube fail in a boiler to which such a valve is attached the operation of the other boilers is not affected, and nothing need be done except to allow the disabled boiler to empty itself.

TABLE 62

DIAMETER AND DRILLING TEMPLET FOR EXTRA HEAVY PIPE FLANGES

MASTER STEAMFITTERS' STANDARD

Diameter of Pipe Inches	Diameter of Flanges Inches	Bolt Circle Inches	Number of Bolts	Diameter of Bolts Inches	Length of Bolts Inches
1	4½	3¼	4	½	2
1¼	5	3¾	4	½	2¼
1½	6	4½	4	⅝	2½
2	6½	5	4	⅝	2½
2½	7½	5⅞	4	¾	3
3	8¼	6⅝	8	⅝	3
3½	9	7¼	8	⅝	3¼
4	10	7⅞	8	¾	3½
4½	10½	8½	8	¾	3½
5	11	9¼	8	¾	3¾
6	12½	10⅝	12	¾	4
7	14	11⅞	12	⅞	4
8	15	13	12	⅞	4¼
9	16	14	12	⅞	4½
10	17½	15¼	16	⅞	4¾
12	20	17¾	16	⅞	5
14	22½	20	20	⅞	5¼
15	23½	21	20	1	5½
16	25	22½	20	1	5¾
18	27	24½	24	1	6
20	29½	26¾	24	1⅛	6¼
22	31½	28¾	28	1⅛	6½
24	34	31¼	28	1⅛	6¾

Note—Flanges, flanged fittings, valves, etc., are drilled in multiples of four, so that fittings may be made to face in any quarter and holes straddle center line.

FIRST NATIONAL BANK BUILDING, CHICAGO, ILL., OPERATING 1,875 H. P. OF STIRLING BOILERS

Boiler and Steam Pipe Coverings

When saturated steam is conveyed through pipes a portion will condense, the amount depending upon the temperature of the steam, and the velocity and temperature of the air surrounding the pipe. This condensation causes a loss not only of volume of steam, but of efficiency in utilizing the remainder of the steam when it reaches the engine, consequently where fuel economy is an object all steam pipes, boiler steam drums, receivers, etc,. should be covered with some efficient heat insulating material, and the saving thus effected will pay large interest on the investment.

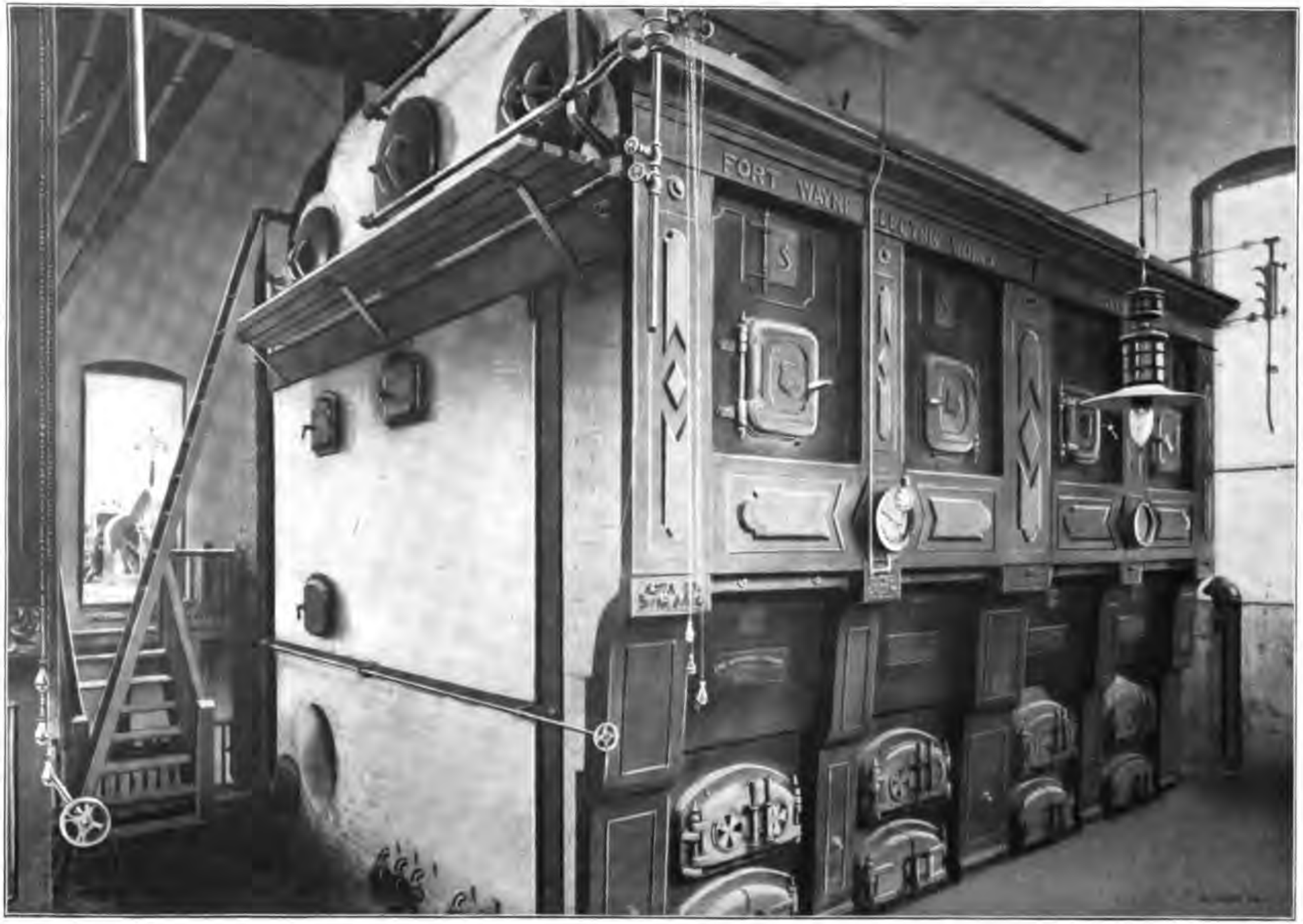
It has been experimentally determined that each square foot of bare iron pipe surface will radiate about 3 British thermal units per hour for each degree Fahr. difference between the temperatures of the steam and the outside air, the exact amount varying with the velocity and humidity of the air. For practical purposes 3 B. T. U. per hour may be assumed. To determine the money value of the loss in any particular case, determine the square feet of exposed pipe surface; determine the temperature of the steam, by referring to Steam Table, page 74, assume the average temperature of the air surrounding the pipe, and then compute the temperature difference. Conditions of operation of the plant will approximately determine the number of hours per annum during which steam will be in the pipe line, hence the total B. T. U's lost per annum can be roughly determined. This divided by 965.8 will give the number of pounds of steam from and at 212° equivalent to this loss; the evaporation per pound of coal and the cost of coal per ton (including cost of handling it and ashes) being known, the money value of the loss is at once derived. Or expressing the matter in a formula:

TABLE 63

EXPERIMENTS ON STEAM PIPE COVERING*

Kind of Covering.	Diam. of Test Pipe. Inches.	Thickness of Covering Inches.	Temperatures Fah.		B. T. U. per Hour per Square Foot of Pipe Surface.		Date of Test.	Testing Expert.
			Steam.	Air.	Total.	Per Degree Difference.		
Hair Felt	2	0.96	302.8	71.4	89.6	0.387	1901	Jacobus
" "	8	0.82	348.3	69.0	117.9	0.422	1894	Brill
Remanit for intermediate pressure	2	0.88	304.5	73.3	100.3	0.434	1901	Jacobus
" " high pressure	2	1.30	306.6	76.1	83.7	0.363	1901	Jacobus
Mineral Wool	8	1.30	344.1	58.3	81.3	0.284	1894	Brill
Champion Mineral Wool	8	1.44	346.1	74.3	86.1	0.317	1894	Brill
Rock Wool	8	1.60	344.1	63.0	72.0	0.256	1894	Brill
Asbestos Sponge Felted	2	1.125	364.8	60.7	145.0	0.477	1901	Barrus
" " "	10	1.375	364.8	62.8	85.0	0.248	1901	Barrus
" " "	2	1.14	309.2	79.4	59.7	0.260	1901	Jacobus
Magnesia	4	1.12	388.0	72.0	147.0	0.465	1896	Norton
"	2	1.09	354.7	80.1	155.8	0.567	1896	Paulding
"	8	1.25	344.1	66.3	106.6	0.384	1895	Brill
"	2	1.08	310.9	81.6	69.8	0.304	1901	Jacobus
"	2	1.00	365.2	64.6	155.0	0.515	1901	Barrus
"	10	1.19	365.2	66.0	103.0	0.347	1901	Barrus
Asbestos, Navy Brand	2	1.20	309.2	79.4	69.9	0.304	1901	Jacobus
" " "	2	1.125	365.2	64.6	176.0	0.585	1901	Barrus
" " "	10	1.375	365.2	66.8	112.0	0.375	1901	Barrus
Manville Sectional	8	1.70	345.5	78.3	93.4	0.394	1894	Brill
" "	2	1.31	354.7	80.1	157.0	0.572	1896	Paulding
" "	4	1.25	388.0	72.0	143.0	0.453	1896	Norton
Asbestos Air Cell	4	1.12	388.0	72.0	166.0	0.525	1896	Norton
" " "	2	.96	303.3	72.3	165.5	0.716	1901	Jacobus
Asbestos Fire Felt	8	1.30	344.7	79.0	133.5	0.502	1894	Brill
" " "	2	1.00	354.7	80.1	198.0	0.721	1896	Paulding
" " "	2	.99	307.4	72.5	180.0	0.766	1901	Jacobus
Fossil Meal	8	.75	347.1	75.3	238.0	0.876	1894	Brill
Riley Cement	8	.75	347.9	74.3	260.0	0.950	1894	Brill

*Arranged from data given in Paulding's *Condensation of Steam in Covered and Bare Pipes.*



FORT WAYNE ELECTRIC WORKS, FORT WAYNE, IND., OPERATING 1,000 H. P. OF STIRLING BOILERS

Cost per annum of steam condensed=

$$\frac{3A(t-t_1)N\times C}{965.8\times 2{,}000E} \qquad [54]$$

In which A=area of exposed pipe surface in square feet.

t=temperature Fah. of steam.

t_1=average temperature Fah. of surrounding air.

N=hours per annum steam is in the pipe line.

E=evaporation from and at 212° Fah. per lb. of coal.

C=cost of coal and handling, per ton.

For long tons the constant 2,000 should be changed to 2,240.

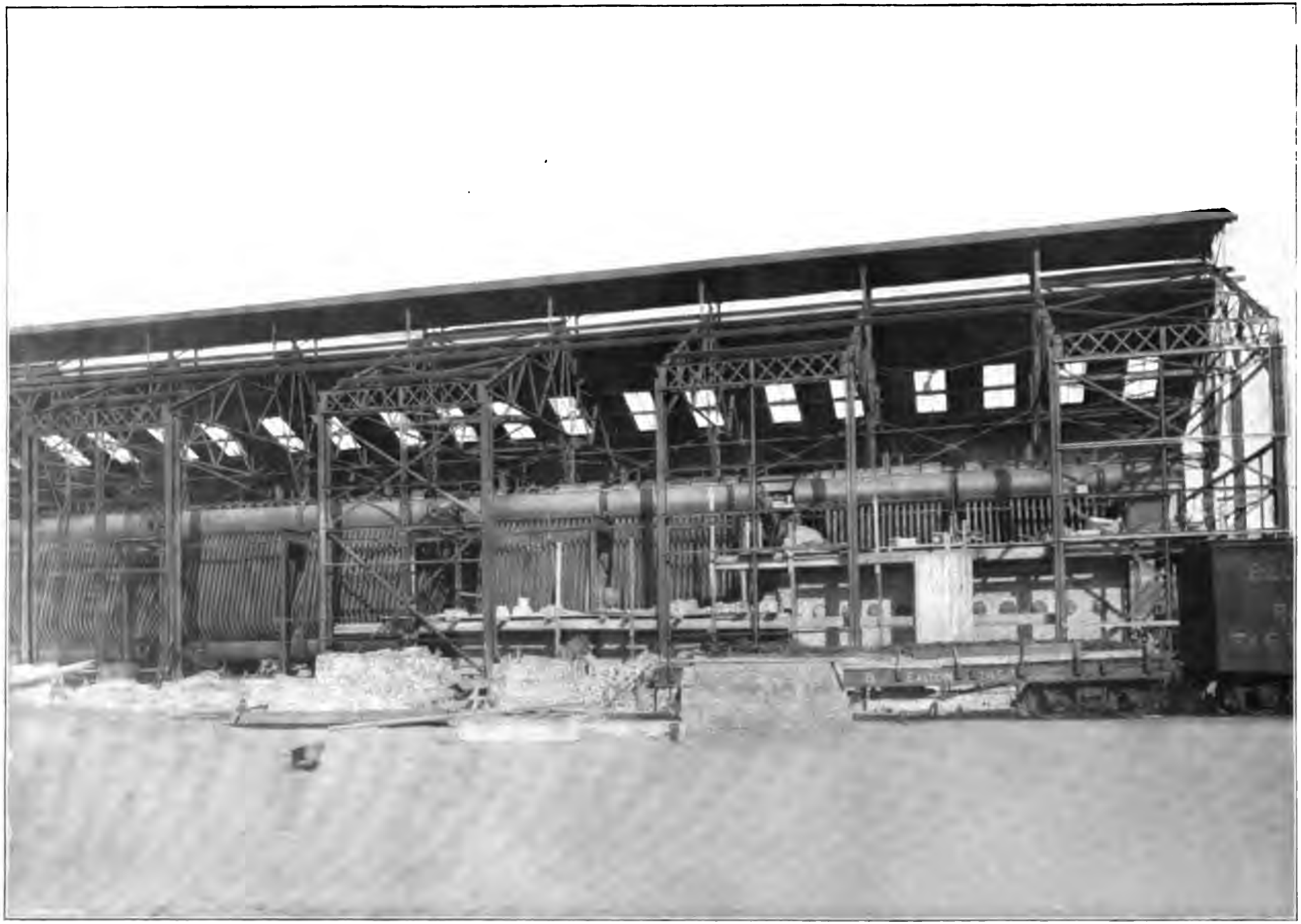
By properly applying a covering of good grade, as much as 90 per cent. of this may be saved. There are many brands of covering on the market, and the only practical way to be sure of what each will do is either to purchase of a firm of established integrity, or else make comparative tests. If the coefficient of conductivity of a covering is known, the heat loss for any given set of conditions can be calculated, but the computation is tedious and the results are only approximate. The method of using this coefficient, and of determining it by experiment, may be found in Paulding's *Condensation of Steam in Covered and Bare Pipes*, together with curves showing the relation between thickness of covering and the heat loss, and other interesting information on the transmission of heat.

It is questionable, however, whether intricate calculations of the heat loss from pipes are of great practical utility, owing to the impossibility of assigning sufficiently close values to the factors which affect all such calculations. About all that can be done is to determine with a fair degree of approximation the relative amount of heat lost by bare and covered pipes. Table 63 has been compiled from tests made by various authorities, and for each covering the B. T. U.'s transmitted per square foot of pipe surface per hour per degree difference between the steam and air temperatures have been computed and entered in the table. Since bare pipe loses 3 B. T. U. per hour per square foot per degree difference in temperature, the per cent. of heat saved by the covering can be computed with sufficient accuracy for all practical purposes. In case of an unknown covering the heat loss per degree of difference can be determined experimentally.

There is a dearth of accurate data on the life of pipe coverings of different kinds. It is well known, however, that as a result of constant vibrations, some of them when on horizontal pipes lose their shape, hang loose on the pipe, and allow the material to shift so that the covering becomes thicker on the bottom than on the top. Only previous experience or careful inquiry as to the experience of others can indicate what defects of this nature may develop in a covering after it has long been at work.

Pipe coverings may be either "sectional" that is, moulded to shape, and attached to pipes by bands, etc., so it can be removed at any time, or "plastic" which is mixed in shape of mortar, and built up on the pipe in layers, so that it cannot be removed and replaced without working it over. The former has more joints, and often under vibration changes shape, but is more convenient for work subject to future alteration. The plastic covering obviates joints, adheres closely to the pipe if of proper quality and workmanship, needs few repairs, and the thickness can be varied to suit. It is more difficult to apply than sectional covering but more permanent when applied.

Pipe coverings should receive the same care and frequent inspection as any other part of a plant. Their efficiency quickly falls off if air is allowed to circulate between them and the pipe, and if allowed to become wet they only increase the evil they are expected to remedy.



PART OF 44,000 H. P. OF STIRLING BOILERS OPERATED BY ILLINOIS STEEL COMPANY, CHICAGO, ILL.

Boiler Cleaning

No boiler can maintain its efficiency unless it is kept clean both inside and out. Deposits of solid matter on the water side of the heating surface interfere with the heat transmission, and cause the metal to burn, blister, or crack, thereby greatly increasing the repair and fuel bills. Systematic boiler cleaning should therefore be included as a regular feature of the operation of every steam plant, and for the purpose of assisting those who do this work the following suggestions are offered.

Cutting Boilers out of Service Preparatory to Cleaning—A boiler should never be emptied while the brickwork is hot; if this be done there will be danger of overheating the metal, and the incrustation will bake to such hardness as greatly to increase the difficulty and expense of removing it.

The best procedure is to allow the boiler to stand at least twelve hours, and more if possible, after the fires are drawn, then to empty it slowly; the manhole plates should then be removed, and the interior be thoroughly washed out with cold water, after which the use of the turbine cleaner should be started. If this plan requires a longer time than is available, the next best method is to allow the boiler to stand three or four hours after the fires are drawn and the main stop valve is closed; the pressure should then be let off, the blow-off valves be opened slightly and water be pumped in at the same rate as it is escaping from the blow-off, which can be done by regulating the pump to the speed necessary to hold the water at the same level in the gauge glass. Pumping should be continued until the boiler is cooled to a temperature low enough to permit it to be opened and washed with cold water, and the use of the cleaner should then begin.

To hasten the cooling of the boiler it is not unusual to open the fire and ash-pit doors and the damper, thus causing a rush of cold air through the setting. While this assists in the cooling, it is an unsuspected cause of the destruction of many furnace walls. A rush of cold air over highly heated fire-brick causes a rapid cooling of the exposed surfaces, uneven shrinkage, and cracking or "spalling" of the brick and loosening of the joints. The degree to which these evil effects are caused by cold air is not generally recognized, and it is advisable *never to throw open the damper and draft-doors until about four hours after the fires are drawn.*

Before starting to remove a manhole plate the operator should, by trying the gauge cocks, or by opening the valve on the steam hose connection, definitely ascertain that there is neither a steam pressure nor a vacuum inside of the boiler. A disregard of this simple precaution has been responsible for many serious accidents to those engaged in cleaning boilers.

Cleaning the Interior—Practically all waters available for boiler feeding contain some impurities which deposit either as mud or as scale. When mud is present it is nearly always precipitated into the lower drum of the Stirling boiler, whence it should be blown out at regular intervals which must be determined by close observation in each case. Occasionally some mud adheres to the rear bank of tubes, and this should be regularly removed by washing down with water applied under a good pressure through a hose terminating in a rose-shaped nozzle.

Scale varies from a porous texture which adheres loosely to the metal, to a hard flinty structure which can be removed only by chiseling or cutting it. Its removal is often tedious but the task should never be slighted Various ways of removing scale have been tried, but experience shows that the most satisfactory method it to use some mechanical device, operated by power, and so designed as to cut or break the scale. Such devices may be divided into two classes: (1) Those which, by very rapid hammer blows, detach the scale by cracking it; (2) those which, after the manner of an emery wheel dresser, cut the scale into small pieces. Tools of the first-named kind have the serious disadvantage of swaging the tubes to a larger or irregular diameter, producing crystallization in the metal, and causing leaks where the tubes are expanded into the sheets, hence

CANDLER INVESTMENT CO.'S BUILDING, ATLANTA, GA., OPERATING 600 H. P. OF STIRLING BOILERS

their use is not to be recommended. Tools of the second-named class are preferable, and of these the most satisfactory is the hydraulic turbine tube-cleaner.

Hydraulic Turbine Tube-Cleaner—This is made in many designs, one of which is shown in Fig. 45. The cylindrical casing contains a hydraulic turbine, consisting of a fixed guide plate which directs the water at the proper angle upon the vanes of the rotating wheel. The water is supplied through a wire-wound rubber hose attached to the upper end of the casing, and the power generated in the turbine is transmitted, through a universal joint, to a cross-shaped head which carries four pivoted arms to the extreme end of which the cutters are attached. This construction makes the tool perfectly flexible, so that with equal facility it will pass through either a straight or curved tube. The thrust on the working parts is taken upon ball bearings or on hardened steel rings, placed between the turbine wheel and the guide plate.

The cutters used under normal conditions are hardened steel toothed disks, similar to those used for emery wheel dressers, as shown in Fig. 46. In special cases other forms of cutters are used, as later described.

When the turbine is revolving at a high speed, the pivoted arms are thrown out, and the cutters, through centrifugal force, bear upon the surfaces to be cleaned and chip away the scale in small pieces. The stream of water flowing from the turbine envelops the cutters, keeps their edges cool, and washes away the scale as fast as it is detached.

In arid countries, where water is scarce, the overflow from the turbine may be run into a sump or sunken tank, in which it can settle and be used repeatedly.

Piping for Supplying the Water—In many cases the turbine is operated by two men, one in the boiler, and another outside to turn the water on and off as required. This method not only requires one man more than is necessary, but causes waste of time in giving orders to operate the water valve. A much better plan is to locate along the rear of the boilers a pipe conveying the water, and to provide this pipe with branches opposite the middle of each aisle between boilers. On these branches place a plug valve, upon the stem of which an S-shaped handle about 18 inches long can be placed whenever cleaning is to be done. A piece of light rope tied to each end of this handle is extended into the boiler drums, and by pulling these ropes the operator regulates the water to suit himself, wastes no time in giving orders, and but one man is required. The pipe supplying the water should be free from reducers, and the hose connection should be full size

FIG. 45. HYDRAULIC TURBINE TUBE CLEANER

Operating the Cleaner—Always use the largest size of turbine which will pass through the tubes. When cleaning begins the turbine should be inserted into the end of a tube, and the operator should have a firm grasp on the hose within 6 or 8 inches of the tube end. When the water is turned on, the rotating parts begin to revolve, the cutter arms fly out, and the attack on the scale begins. The operator should then immediately begin moving the turbine alternately up and down and continue this as long as the tool is working. The water should not be cut off while the

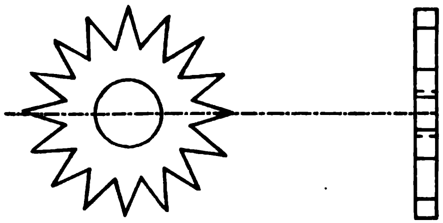


FIG. 46. TOOTH DISK CUTTER

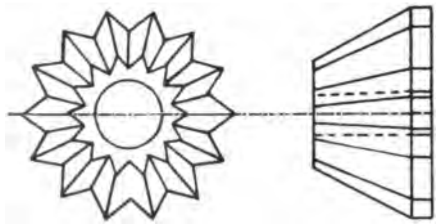


FIG. 47. CONOIDAL CUTTER

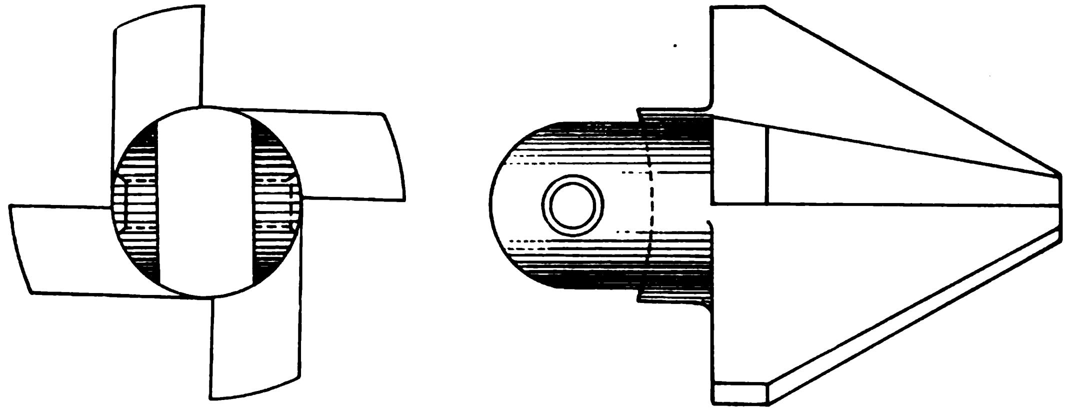


FIG. 49. DRILL HEAD FOR HEAVY SCALE

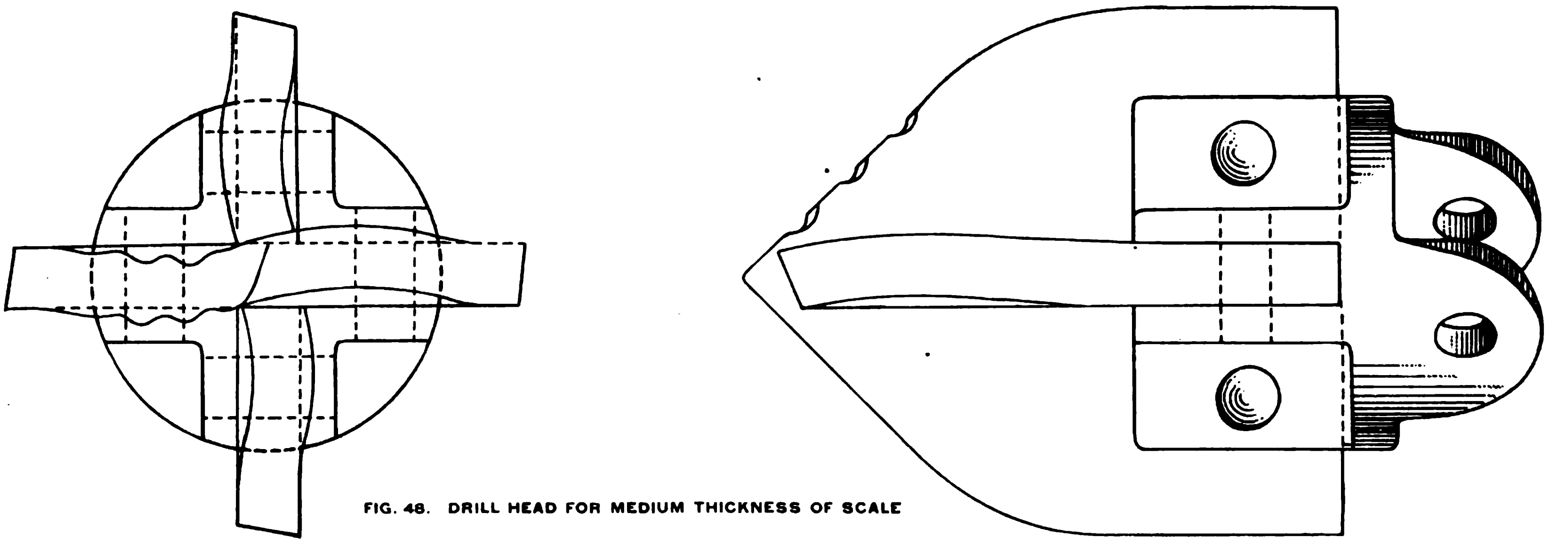


FIG. 48. DRILL HEAD FOR MEDIUM THICKNESS OF SCALE

FULL SIZE DETAIL DRAWINGS OF CUTTERS FOR USE IN CONNECTION WITH THE TURBINE TUBE CLEANER

cleaner is in the tube, nor should the cleaner be permitted to remain stationary — it must be kept constantly moving up and down. As the scale is removed, the turbine will gradually travel farther down the tube, and when it finally reaches the mud drum the tube will be clean, unless the tool has been either pushed through too fast, or allowed to draw itself through. Usually the weight of the tool and hose will tend to draw the turbine down, and if allowed to go ahead too fast, it may remove the scale from a spiral-shaped strip, leaving it in other places, hence it is important that the operator train himself to detect by the sound, and vibration of the hose, the character of the surface upon which the cleaner is working, and to regulate the forward motion of the cleaner accordingly.

The toothed disk cutters give entirely satisfactory results when the scale does not exceed $\frac{1}{8}$-inch in thickness, which covers all cases of normal use, since except in an emergency, scale exceeding $\frac{1}{8}$ to $\frac{3}{16}$-inch thick should never be allowed to form in the rear bank of tubes between regular cleanings, and $\frac{1}{16}$-inch thickness of scale on tubes in the front or middle bank is *prima facie* evidence that the boiler has been neglected. In cases of neglect, not only will a greater thickness of scale form, but by long exposure to heat it bakes harder than otherwise, which greatly increases the difficulty and expense of removing it. As soon as a boiler is found to be heavily scaled, it should at once be cut out of service and cleaned. If it is found that the scale does not exceed $\frac{3}{8}$-inch thick, the toothed disk cutters should be replaced by solid conoidal cutters, made of tool steel, as in Fig. 47. Place these with the small end downward.

Should the scale exceed $\frac{3}{8}$-inch in thickness, it is best removed by a four-lipped drill, made of hardened tool steel, with cutting edges at an angle of 45 degrees to the axis of the tool, as in Fig. 48. When the scale is excessively thick, it should first be attacked by using a drill head of the form shown in Fig 49. which should be followed by Fig. 48.

The tools represented in Figs. 48 and 49 are fastened to the turbine by removing the cross-shaped piece which carries the arms, and substituting the head to which the drill is attached. When using these tools on heavy scale the turbine should be handled with judgment, as it is evident that the tool cannot be advanced as rapidly as when the scale is thin.

By careful observance of the foregoing instructions it will be possible to remove any deposit found in a tube, but the time required will of course vary according to the nature of the scale, and the thickness to which it has been allowed to accumulate.

Care of the Turbine Cleaner—Those who have had no experience with turbine cleaners are advised to take the tool apart, and familiarize themselves with its construction. While the tool is well built, it must be properly cared for if satisfactory results are to be obtained. When each boiler cleaning is finished, the turbine should be thoroughly washed, then stored in a pail of oil.

How to Soften Refractory Scale—When scale has been allowed to accumulate and bake to an excessive hardness its removal is difficult, The work may be expedited by introduction of some agent which will rot and soften the scale. One method is to introduce 40 to 80 lbs. of carbonate of soda (ordinary soda-ash) according to size of the boiler, block the safety valves open, then allow the water to simmer gently; in a particularly bad case this may require several day's boiling. The scale will thus be softened so that it can easily be cut; the boiler should finally be thoroughly rinsed with fresh water, or foaming may result.

Kerosene is often used for the same purpose. Some spray it over the surface with a squirt pump, or other means, while some fill the boiler nearly full of water, introduce a quantity of kerosene, then open the blow-off valve very slightly, so that as the water level slowly falls the kerosene is brought into contact with every portion of the interior surface. Before one enters the boiler it should be thoroughly ventilated to remove volatile gases from the oil, since they are highly explosive.

Kerosene should never be used without first testing it for free acid which is liable to be present from the refining. Insert blue litmus paper, and if it turns red the oil should be rejected since the acid will cause corrosion.

Oil in Boilers—Scale is not the only cause of burned tubes. Burnt and blistered tubes will be the inevitable result of allowing oil to enter boilers. When the presence of oil is detected the boiler should at once be cut out of service, and as soon as it has cooled it should be emptied, and several pailfuls of soda-ash be placed into the mud drum. The boiler should then be filled with water, and be gently fired for 12 to 15 hours, keeping the steam pressure down to 12 or 15 pounds. At intervals, a portion of the water may be let out, and be replaced by pumping in an equivalent amount; finally the boiler should be allowed to cool, then be emptied and the interior be thoroughly rinsed with fresh water.

Cleaning the Fire Side of the Heating Surface—The fire side of the heating surface should be kept clean, and under no circumstances should a thickness of soot exceeding $\frac{1}{16}$ of an inch be allowed to form, or the boiler efficiency will be greatly decreased. The surface of the tubes and drums should be regularly blown off with steam by using the hose and steam blower-pipe furnished with the boiler. Since the heating surface can be blown off in 10 to 15 minutes, there is no reason for neglecting this work. The interval between such cleanings will depend upon the character of the fuel; with smoky fuels the cleaning should be done *at least* once per day, preferably during the period of lightest load; the surface should also be well brushed off at regular intervals.

Never use the steam-blower when the boiler is cold, or the steam will condense, dampen the brickwork, and cause the sooty deposits to become gummy and adhere to the metal. It is advisable to have not less than 50 pounds pressure on the boiler when the steam-blower is used, and there should be a fire on the grates to insure the brickwork being hot.

Necessity for Periodical Examinations—No matter how excellent the feedwater may appear to be, every boiler should at regular intervals be examined, and even though no trace of deposit be noted the cleaner should be passed through the tubes, to ascertain their condition, and enable one to *know* that they are clean.

The mud drum should be inspected regularly, and all accumulation of soot, ashes, or dirt, be thoroughly removed. In all cases where coal, particularly anthracite, is used, the accumulations on the mud drum should be blown off before the boiler is emptied. The reason for this is that such deposits often contain coal which holds fire for many hours after the furnace fires are drawn, and the heat due to this cause may damage the drum or tube ends if the boiler is emptied.

All soot, ashes and dirt should be removed from the top of furnace arches every time the boiler is cleaned. The top of the bridge wall should be swept clean, and any defects in the brickwork should be repaired. The asbestos rope joint between brickwork and ends of the mud drum should be inspected, and unless found perfectly air tight, it should be made so before the boiler goes into service. The opening where the blow-off pipe passes through the wall should be inspected, and if air leaks are found they should be stopped by caulking with asbestos rope. All cleaning doors should be examined, and if air leaks are detected they should be stopped.

Whenever a boiler is off for cleaning it is an excellent plan to make a careful general inspection of it both inside and out, including the setting, cleaning doors, valves fittings, etc. Incipient defects may thus be discovered and corrected at a trifling expense, serious troubles will be obviated, and the general operation of the plant will be improved. It is doubtful whether the short time necessary for such inspection could be more profitably employed, since a boiler plant continually proves the truth of the ancient maxim, "A stitch in time saves nine."

Care and Management of the Stirling Boiler

Before starting a new Boiler—Make a complete examination of boiler and setting. Inspect all valves, fittings and attachments, and ascertain that they are properly connected and in perfect order. See that nuts on all tie rods in frame are turned up tight, and at intervals during the first six months after the boiler is in operation, ascertain if these nuts are tight, and if not, make them so.

Inspect the brickwork, and see that height of fire-arches and distance between end of fire-arches and nearest tube corresponds with drawings. *See that the mud drum can expand freely;* that baffle openings are as marked on drawings, baffle tiles all in place and joints sealed with fire clay, and that all mortar and rubbish which has dropped down on the mud drum while brickwork was in progress, has been scraped off and drum surface brushed clean. *This is frequently overlooked.* See that blow-off pipe and valve, as well as blow-off main to which they are connected, can move freely; *on no account should they be cemented or bricked in, but should lie free in a trench or slot.* See that space around the blow-off pipe where it passes through rear setting wall is plugged with asbestos rope until it is air-tight.

When oil or gas fuel is used, take particular care to ascertain: if the fire arches terminate at proper distance from the tubes; checkerwork walls in rear of grates are of proper height; openings between brick over grates are of proper width, and burners set to blow the flame parallel to the brick over grates and at proper distance above them; to minimize effect of gas explosions, turn back the latches of cleaning doors, so the doors can blow open freely and release the pressure due to the explosion.

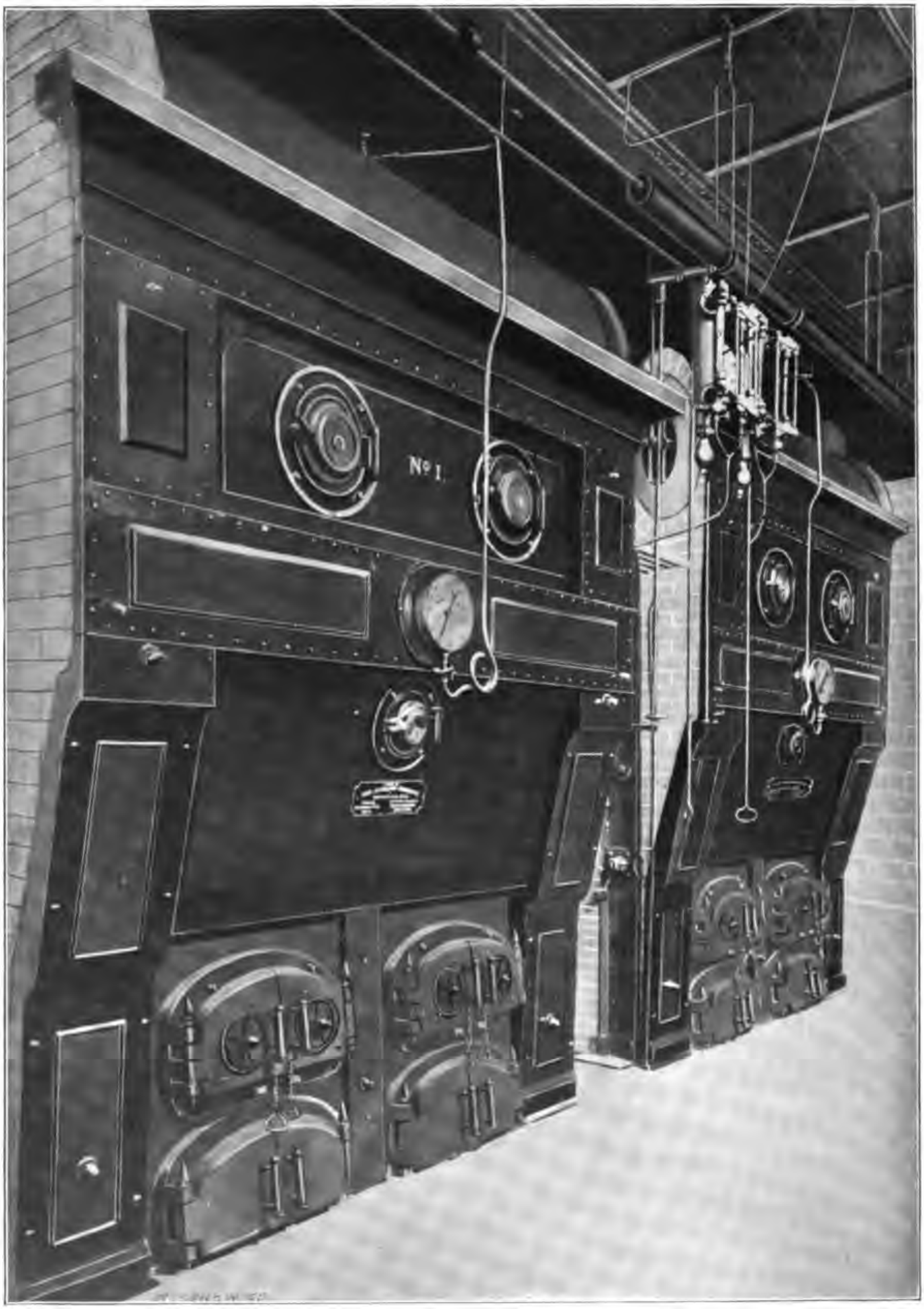
See that all dirt, waste, and tools, are removed from interior of boiler. Then place in boiler about a peck of soda-ash, fill to usual level with water, boil gently for several hours after boiler is finally heated up, let stand till cold, then empty, and finally give interior of boiler a thorough washing with cold water. This will remove all oil and grease and prevent foaming when the boiler goes into commission.

Firing a Boiler with Green Walls will invariably crack the Setting, hence it is absolutely necessary to dry out the brickwork properly. If circumstances permit, it is advisable as soon as stack connections are made, to block open the damper and ash-pit doors, so that circulation of air will aid in drying the brickwork. The next step is to fill the boiler with water and put in a light fire of shavings, which may gradually be increased by using some wood, continuing until the walls are thoroughly dried inside and out. This will require several days, but by close observation the walls, if built according to instructions (page 233) can be dried out without cracking.

When steam is available an excellent method of drying the brickwork is to connect temporarily a small steam supply pipe to the new boiler, and to attach a trap or other drainage apparatus to the blow-off pipe. The new boiler when filled with steam will then act as a large radiator, and will heat the air around it, hence if ash-pit doors and damper be left open there will be a steady current of warm air passing through the setting, and the brickwork will be gradually and effectively dried out. The steam supply should be very small at first, and be increased as the drying-out proceeds.

Cutting Boiler into Steam Main—Under no circumstances whatever should a boiler be "cut in" with other boilers unless the pressure within it is identical with that in the main. Before opening the boiler stop valve or the header valve, *be sure* that there is no water in the length of pipe between these two valves. Steam valves should always be opened or closed *very slowly*, and the valves should first be eased from their seats slightly for some moments to permit a circulation to become established before the valves are fully opened.

When the boiler is in service, observe the following carefully and at *regular intervals* make an inspection so you will *know* everything is right.



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Steam Gauges—When pressure is off these should stand at zero, and when safety valve blows off, the gauge should indicate the same pressure for which the valve was set to pop. If it does not, one is wrong, and the gauge should at once be compared with one of known accuracy, and any error be rectified.

Safety Valves—These are useless if allowed to become stuck on their seats. To prevent this, cause the valves to pop once on *every shift*.

Water Level—Never fire a boiler without first ascertaining that it contains the necessary quantity of water. When operating, never depend on gauge glass, or water alarms alone, but try the gauge cocks.

Gauge Cocks, Water Gauges, and passages to gauge must be kept clean, and should be blown out frequently. Formation of colored rings on the glass, due to oil or other deposits, is misleading, and should be obviated by frequent cleaning. Automatic water gauges of all types and all other automatic appliances, need frequent inspection.

Blow-Off Valves must be kept tight, and should be *known* to be tight. Every blow-off pipe should be so arranged that a leak can be seen. This can be arranged by placing in the pipe just beyond the blow-off valve, a tee with a 1″ outlet to which a gate valve is attached; this valve should be left open except when blowing-off is in progress, so that it will act as a tell-tale in case the blow-off is leaking. Where boiler water deposits material that is liable to cut the blow-off valve and cause leaks, a gate or asbestos packed plug cock should be placed between boiler and regular blow-off, so it can be closed and permit the blow-off valve to be cleaned without shutting down the boiler.

Firing—The method of firing coal to be adopted will depend upon the kind of coal used; the chapter on Fuel Burning should therefore be carefully studied, and the most efficient method of firing be determined by close observation and experiment. The draft should be regulated to the least amount necessary to maintain the desired rate of combustion.

When burning wood, carry as thick a fire as the draft will allow; the fresh wood on top tends to force down the partly burned wood thus covering the grate with a bed of coals, and reducing the air excess.

When burning oil, see that the flame is white, not red, and free of sparks which indicate incomplete combustion. The air supply should be regulated to a point where further diminution of it will cause smoke to appear in the stack. *Avoid squirting unatomized oil on the tubes* as each spot where it touches is liable to develop a blister.

Foaming—If caused by excessive demand for steam, checking the outflow of steam will usually stop it. If caused by excess of dirt due to concentration, blowing down and pumping in clean water will usually stop it. In case of violent foaming, check the draft and fires. The Stirling boiler never foams with good water unless the water is carried too high, in which case lower the water-line, which should never be carried higher than two gauges when the boiler is steaming.

Blowing Off—When feed water is salty or muddy, blow off a portion at as frequent intervals as the conditions demand. Empty the boiler every week or two and fill up afresh, but *never empty the boiler while brickwork is hot, and never feed cold water into a hot boiler*. *Always* blow off all accumulations of soot, fuel, etc., from the mud drum before emptying a boiler. The fine coal, particularly anthracite, carried over on the drum by the draft, may hold fire many hours after the furnace fires are out. If under such conditions the boiler is emptied the mud drum and tube ends may be overheated, and leaks or broken tubes are liable to result.

Low Water—Immediately cover fire with ashes or earth, preferably wet; in default of anything else handy use fresh coal; the important point is to check the heat as quickly as possible. Draw the fire as soon as it can be done without increasing the heat. Do not turn on the feed, lift safety valve, start or stop engine, until boiler is cooled down. Before firing the boiler again, put on the cold water test, locate any leaks, and stop them.

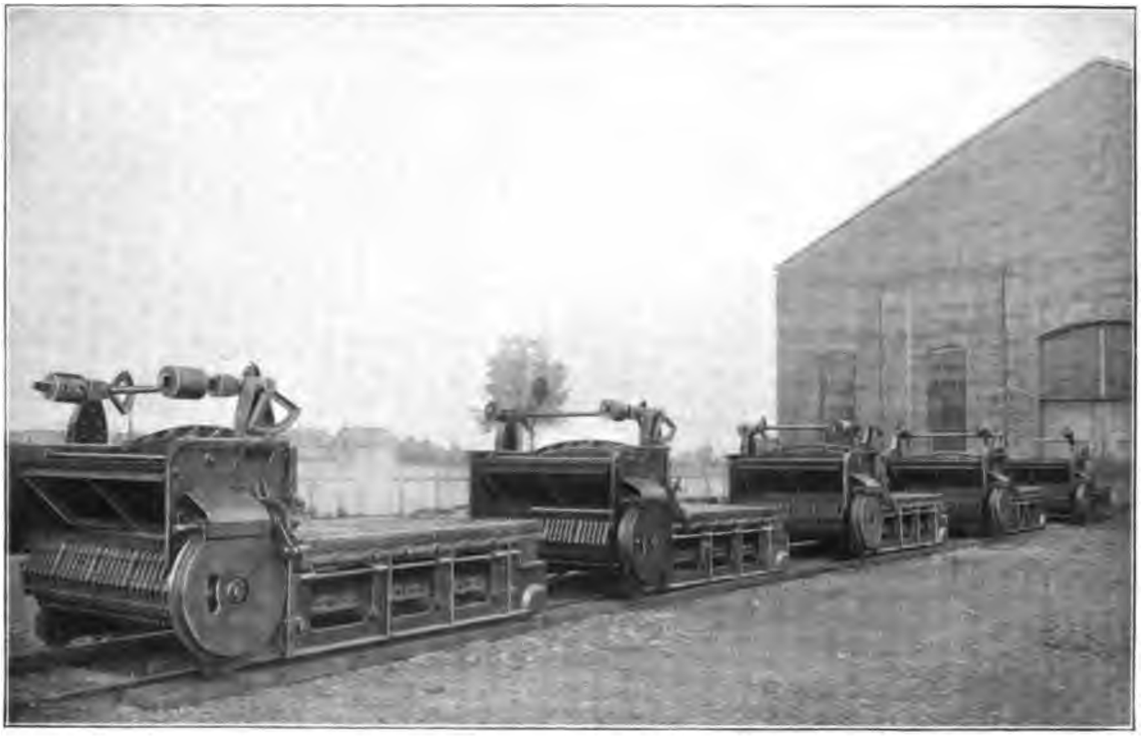
Cleaning—To avoid waste of fuel and deterioration of the boiler, keep it clean inside and out, by carefully following the directions given in chapter on Boiler Cleaning.

Oil in Boilers—Burnt and blistered tubes will be the inevitable result of allowing oil

to enter boilers. As soon as the presence of oil is detected the boiler should be cut out of service, and be treated as directed in chapter on Boiler Cleaning.

Air Leaks—All air that enters boiler or breeching except by passing through the fire, causes losses which are often large and unsuspected. Carefully test for leaks around cleaning door frames, blow-off pipes, dampers, breeching connections, etc., and carefully plug each with asbestos rope or cement.

use unless it receives proper attention as soon as its use is discontinued, hence the following instructions must be carefully observed. Before emptying the boiler place in each upper drum several gallons of crude oil so that when the blow-off is opened the oil will form a light covering over the inside surface of all tubes and drums. (Before the boiler is started again, remove this oil with soda-ash as already directed.) Dry the boiler thoroughly when emptied out.



STIRLING CHAIN GRATE STOKERS READY FOR SHIPMENT

Steam or Water Leaks should be stopped without delay, and extra precaution should be taken to exclude water from those portions of the boiler covered by brickwork, otherwise unsuspected corrosion may occur. Leaks in steam pipes over the boilers should be located and stopped.

Internal Corrosion of Boiler—This is caused by some harmful agent in the water, and the matter should at once be referred to some competent chemist experienced in investigating boiler feed water.

Standing Unused—If a boiler remains idle it will *deteriorate much faster than when in*

If the boiler cannot be emptied, fill it quite full of water, to which has been added a quantity of soda-ash, then boil off the air and close the boiler air-tight.

Remove the baffle tiles, thoroughly sweep off all accumulations of ashes and soot with a wire brush, and give all tube and drum surfaces a coat of boiled linseed oil. Smear all brass or finished work with vaseline slush, or a mixture of white lead and tallow.

Cover the stack tops with a water tight hood, and *see that no water* can reach the boiler through breechings, openings in roof, or other sources.

Specifications for Masonry in Stirling Boiler Settings

To secure satisfactory service from the boilers, it is absolutely essential that the setting be constructed with utmost care, and of the best materials. After the setting is completed it should be carefully dried out as directed in chapter on "Care and Management of the Stirling Boiler" (page 229) and from time to time inspection should be made to locate any cracks or loose brickwork, and these should be at once repaired. Prompt attention to this matter will not only insure more efficient results from the boiler, but obviate unnecessary repair bills.

The following specifications should be observed during progress of the work:

Excavation—Consult the drawings and stake off accurately according to dimensions given thereon. Where boilers rest upon rock, excavate for ash-pit and space under mud drum. In other cases excavate to depth shown upon drawing or to such additional depth as is requisite to insure a solid foundation.

Concrete—Cover the entire surface with concrete to a thickness of 8 inches, unless the nature of the soil should require a heavier bed. When found necessary do not hesitate to make the bed heavier. The composition of the concrete should be one yard of rock broken to pass through a 2-inch ring, ½ yard of clean, sharp sand, and 2½ barrels of Portland cement. Clean gravel will answer as well as broken rock. Mix well when dry, then wet and mix thoroughly. Avoid using too much water; 40 pounds of water to each 100 pounds of cement will be more than ample. When placing the bed of concrete on foundation cover only such space as the quantity mixed will bring to the thickness required; work rapidly; ram the concrete well with a 50 lb. rammer having a face 8 inches in diameter, and continue to ram until the mass is so compacted that water appears on the surface. See that the bed of concrete is level and smooth when finished.

When concrete does not cost more than brickwork, the entire foundation up to floor line is frequently made of concrete with excellent results.

Red Brickwork—Carefully follow the drawing in laying off the brickwork and use good, hard, well burned brick, uniform in dimensions. Wet the brick before laying. From the concrete bed to the floor line use a well mixed mortar, composed of one part Portland cement to two parts of clean, sharp sand, grouting each course thoroughly with slush mortar. While the drawings show a cap stone under each support, this is imperative only where the brick has not a crushing resistance equal to eight times the load to be carried. When the cap stone is omitted, it may be replaced by a pier made of concrete, or of selected hard brick carefully laid in Portland cement mortar.

From the time the work is started until the last brick is laid, do not forget the importance of tight walls. See that every brick is properly bedded, and every joint well and thoroughly filled with mortar. Air leaks through the walls ruin the draft and destroy the efficiency of the boiler.

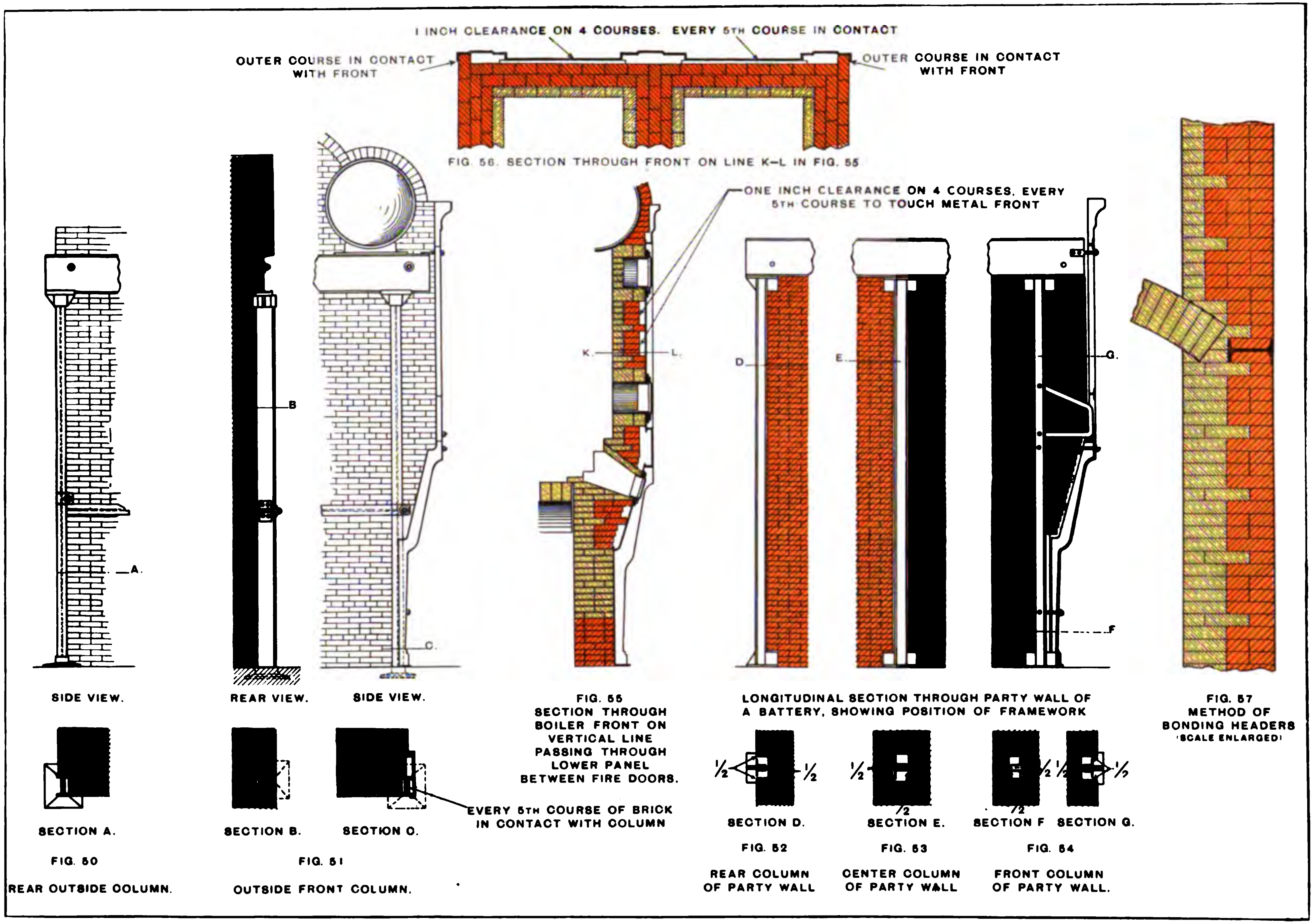
All walls must be built straight and plumb, and carried up simultaneously, thoroughly grouting each course.

On all red brickwork, from the floor line up, use well mixed mortar, composed of one part lime to three parts clean, sharp sand. Mortar must not be used while hot, caused from the slacking of freshly burned lime.

The brick should be laid in courses of four stretchers to one header, *i. e.*, every fifth course should be a header course.

As the work progresses, see that all stay rods and anchor bolts are properly placed; set and anchor all door frames as shown on drawing, turning over each door an arch extending through the thickness of the wall. See that there are no air leaks between the door frames and the brickwork.

The mud drum must be perfectly free, to allow the tubes to expand and contract as the conditions may require, and at no place must the brickwork be allowed to touch either the mud drum or the blow-off pipe. This rule positively admits of no exception. *Nothing must affect the freedom of the mud drum.* At the manhole end of mud drum



DETAIL DRAWINGS SHOWING METHOD OF FITTING BRICKWORK AROUND THE STRUCTURAL STEEL FRAMEWORK SUPPORTING STIRLING BOILERS

turn a complete arch circle, the inside of the circle being kept 1-inch clear of the drum. This space to be afterwards filled with a ring of 1-inch asbestos rope. Before the blow-off valve is attached, slip over blow-off pipe a piece of 4-inch pipe 12 inches long, and build same into wall as a thimble through which blow-off pipe passes, then plug up the annular space between the two pipes with asbestos rope or fiber.

The bridge wall must be carefully laid to allow the mud drum freedom. There must be a space of 1½ inches between the top of the bridge wall and the front row of tubes.

In laying the red brick on top of the steam drums and the steam circulating tubes, use every precaution to have tight joints so as to thoroughly exclude air leaks. These bricks are to be laid in lime mortar, with the joints open at the upper edge, and, after all the bricks are in place, these joints should be slushed with cement, leaving a covering on top of the brick at least ¼-inch thick. *The thinner the joints between the bricks the longer the setting will last.*

Fitting Brickwork Around Boiler Supports—Figs. No. 50 to 57 show how this should be done. The red bricks should be laid up close against the inside flange of the outer columns as shown in Figs. No. 50 and 51. Along the rear side of the front column, and the front side of the rear column, the face of the brickwork should be in line with the outer edge of the flanges of the I-beam column, with the exception that every fifth course of bricks should be placed into contact with the web of the columns, as marked in Fig. 51.

When two boilers are set in battery there will be three angle-iron columns placed inside of the party wall, and all around these columns a clearance of one-half inch must be left between them and the brickwork, as shown in Figs. 52, 53, and 54.

When building the brickwork behind the metal front of the boiler care must be taken to see that it is placed in exact accordance with Fig. 55. The outer course of the side walls should be carried up to, and closely fitted against, the inside of the face of the outer columns and pilasters, so that the flange of the pilaster or column overlaps the side wall as shown in Fig. 56. The remainder of the brickwork behind the metal front should be carried up so that its forward face is one inch behind the panel plates, with the exception that each fifth course of brick should project forward sufficiently to come into contact with the metal front, thus assisting in supporting it and in holding the panel plates securely in place, as illustrated in Fig. 55.

In order that the brickwork may be perfectly tied or bonded, the header courses on the inside and outside faces of the wall should be on the same level and abut each other in the center of the wall; a course of headers must also be placed inside of the wall both above and below the outside header courses, and across their abutting line, as in Fig. 57.

Lime Mortar—Lime is greatly improved by allowing it to stand as long as possible between time of slacking, and using in the wall. In some countries lime is slacked and allowed to remain in pits a year before using, as the first slacking is not complete, and the mass contains small particles which slack only after long standing. When freshly slacked lime is used in a boiler setting, these unslacked particles finally swell, the mortar gets loose and "shattered," and the brickwork is a failure. Lime for boiler setting should be slacked at least six weeks, or longer if possible, before using.

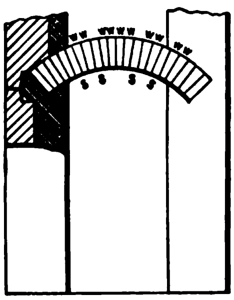
Fire-Clay—Fire-clay is *not a cement*, and it has little or no holding power. Its office is therefore not to act as a binder, but merely to fill the voids. In consequence a fire-brick joint is the more perfect in proportion as the quantity of fire-clay approaches the amount necessary to fill the voids, without preventing the brick from touching, precisely as in case of a glue joint between pieces of wood. Clay of consistency sufficient to permit use of trowel should not be permitted; the proper way is to mix the clay to requisite thinness, dip each brick into the clay, "rub and shove" each brick into final place, then drive it with mallet or hammer and block, until it actually touches the brick below it. *Rigid adherence to these directions is absolutely essential when constructing fire-arches.*

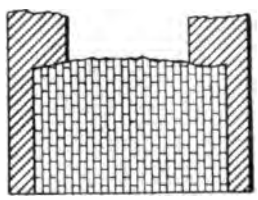
The two defects of fire-clay are its shrinkage during drying, and its lack of cement-

ing power. The former may be greatly diminished by adding to the clay about 20 per cent. of its volume of fire-brick pulverized and sifted to *fire-brick flour*. This can be obtained in many places, but unless it is of the requisite fineness, avoid it, as coarse material will thicken the joints an amount which offsets the advantage.

The cementing power of fire-clay may be increased by adding to and slacking in with it about 1½ per cent. of its volume of lime; measure the clay and for each cubic foot, put in a piece of lime not exceeding 4×2×2½ inches. This will have just sufficient fluxing power to unite with the clay and form a hard clinker which takes a grip on the fire-brick. It should always be used when building arches.

measurements carefully, and lay off the lines of the skewback. When determining length of arch, be sure to consult the drawing for the particular job on which you are working, as the arch length varies with character of fuel. While laying off the skewback, and while laying the brick in same, use a true straight edge the length of the arch. See particularly that the skewbacks have no lumps, bumps or other irregularities and the brick wall behind them is *absolutely solid*, and the red bricks laid close together so there will be no space filled with mortar or spalls. See that the two skewbacks are perfectly parallel, level and in line, the one with the other. Keep using the straight edge until the surfaces are smooth and regular.





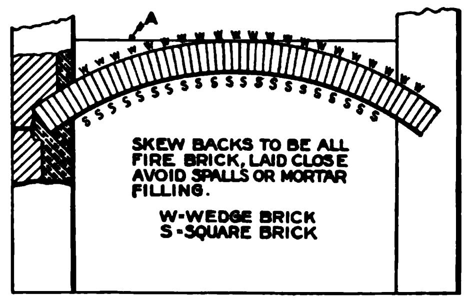


FIG. 58 DETAILS OF FURNACE ARCHES FIG. 59

Fire-Brick Work—When the point in the brickwork has been reached where the fire-brick commences, the fire and red brick must be carried up together, and from a distance 6 inches below the bottom of the grate bars to the skewback of the arch, every course must be a header course, and every fifth course of the fire-brick must be tied into the red brick. The best fire-brick obtainable must be used in the side walls of the furnaces and arches, beginning 6 inches below the bottom of the grate bars, and carried up on both side walls to the top of the arch, and from the front wall back to the front baffle-tiling. The arches must be constructed also of the very best grade of fire-brick obtainable. All fire-brick must be closely laid with a solution of fire-clay and water as above directed.

Fire-Arches—When the arch above the fire is reached, consult the drawing, take

Next, set the center upon which the arch is to be turned and make the center as follows: cut from 1½-inch plank three segments of the proper length and radius. Let the distance between the two outer segments be 6 inches less than the length of the arch; place the third segment in the center of the two outer ones. See that they are parallel with each other and square, so that there will be no wind in the center when nailed up. Batten the segments with 1-inch square strips, laid close together, said strips being smooth and straight.

After the strips are well and securely nailed to segments, plane off to a true circle. When the center is completed, set in place, being careful that the two outside strips line exactly with skewbacks. If they will not line, either the skewbacks or center must be wrong, and the defective one should be righted before a single brick is laid.

When the center is set and found right, select smooth, straight and uniform rectangular bricks and wedge bricks (bull heads); have the solution of fire-clay soft and well mixed. Do not use a trowel; dip the bricks and shove up close, driving to place with a brick hammer, or mallet and block.

Keep the joints as thin as possible.

The bricks must positively *not* be laid in consecutive rings; every joint must be broken and have a bond equal to one-half the width of a brick. While laying the arch, alternate square brick with bull heads and *vice versa*, as may be found necessary to maintain a true circle. Be particularly careful that the arch is not turning too fast. When the keying course is reached, try the bricks in dry and see that they have the proper taper. If not of the proper taper cut them nicely to fill the space.

Do not leave any interstices to be filled with fire-clay, as it will only fall out when dry and let the arch down.

The keying course should be a snug fit, driven carefully to place by laying a small block of wood on top of the brick, on which hammer lightly, being careful not to drive so hard as to crush or otherwise mutilate the brick. When properly keyed, remove the center.

Figures 58 and 59 fully illustrate how this arch should be built. While the figures show the smallest and largest arches, the same general plan of procedure covers all intermediate sizes.

Finally, at end of arch, run a 9-inch wall across the spandrel openings, as at A, Fig. 59, to provide a parallel throat for gases as they pass into the tubes.

Fire Door Arches—For building the fire door arch The Stirling Company furnishes a special skewback. Consult the drawing, and the accompanying cuts, Figure 60, and follow them closely, using the same careful methods advised in building the fire-arches. The bricks (bull heads) in this arch must be turned in consecutive rings, making no attempt to bond the one with the other.

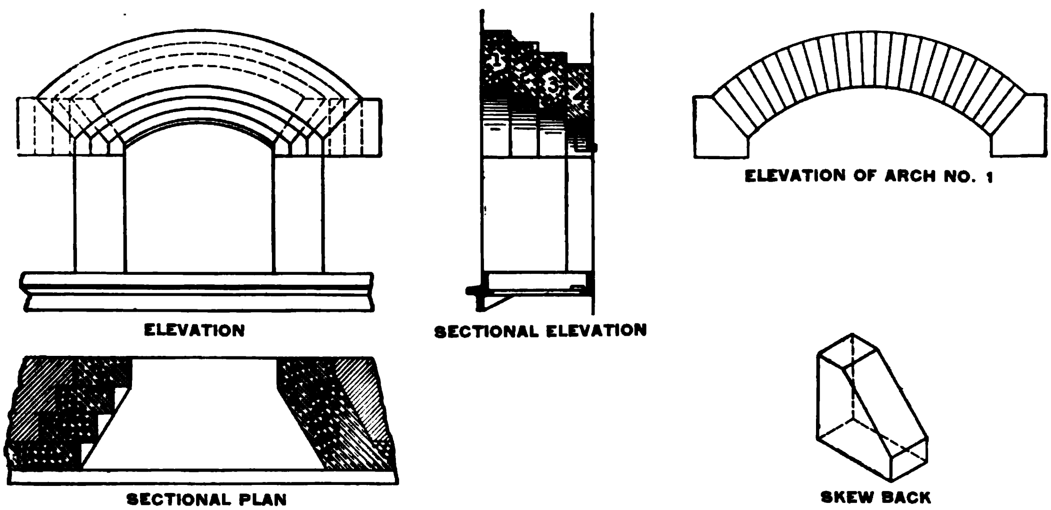


FIG. 60. DETAILS OF FIRE DOOR ARCHES

The fire bricks laid above the water circulating tubes must *not* be laid tight nor keyed like an arch. Dip the bricks in fire-clay and lay snugly to place; when completed grout with fire-clay wash, leaving a good coating above the bricks.

Baffling Tile—Consult the drawing and see that all supports and hangers for the baffling tiles are in proper place. If found wrong have them righted before laying a single tile. Commence at the side wall and lay the tiles in rows longitudinally with the drums, laying a single row at a time.

See that the first tile fits smoothly and closely to the side wall, and that the other edge comes immediately back of the vertical

center of the tube. If found too wide, cut to fit. Care at this point will bring every seam in the row exactly back of the tube centers. Fit the entire row dry, and see that each tile lies closely and snugly to its companion.

Follow all these instructions until all the tiles are laid, watching that the tiles fit closely at the joints, top, bottom and sides. See that there are no bumps on the portion of the tile face that comes into contact with the tubes; let it lie smoothly and evenly in place.

PART OF 1,500 H. P. OF STIRLING BOILERS, WOLVERINE COPPER MINING CO, KEARSARGE, MICH.

Here also, the trowel must be set to one side. Dip the edges of the tiles in a fire-clay solution, then lay them into place carefully.

There must be no leaks in joints nor stopping up of interstices with fire-clay to fall out when dry, and thus divert the gases from their proper course, allowing them to take short cuts to the smokestack.

After all is done, give the joints several coats of clay wash which should be made of a thin solution of fire-clay, and be applied with a whitewash brush.

All cleaning doors are to be located in side and rear walls as shown on blue prints; fit the brick close up against the door frames to prevent air leakage.

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The figures refer to the pages.

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